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Engineering Thermodynamics

Third Edition

P K NAG

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Preface to the Third Edition

The response to the second (previous) edition of this book has been overwhelming with its 25th reprint since 1995 exhausted. *Engineering Thermodynamics* is a basic subject so much so that there is hardly any scope for major revision. Simple precise statements of basic underlying principles have always been emphasized. However, in this edition, certain portions have been updated, some new material like chemical exergy and fuel cell have been added, and a good number of new problems have been included, of which some are solved while others are given as exercise. Chapter 19—Gas Compressors is new chapter that has been compiled by taking away the relevant matter furnished in Chapter 10 of the earlier edition and making additions to it.

The Solution Manual providing solutions to the chapter-end exercises given in the book is available for the teachers.

The author hopes that this new edition will continue to remain a popular book on the subject as it has always been since its initial publication in 1981, about a quarter of a century ago. He would appreciate any suggestion towards improvement or detection of errors from the readers.

P K Nag

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Preface to the First Edition

The purpose of this book is to provide a mature approach to the basic principles of classical thermodynamics which is one of the few core subjects for the undergraduate students of almost all branches of engineering. The system-surroundings interactions involving work and heat transfer with associated property changes, and the system-control volume approaches of the first law have been emphasized. The second law has been elaborated upon in considerable detail. Except for some physical explanations, a statistical or microscopic analysis of the subject has not been made.

The first eight chapters of the book are devoted to a thorough treatment of the basic principles and concepts of classical thermodynamics. The second law and entropy have been introduced using the concept of heat engine. Chapters 9 and 10 present the properties of substances. Chapter 11 gives the general thermodynamic relationships among properties. A detailed analysis of power and refrigeration cycles is given in Chapters 12 to 14. Chapter 15 deals with air-water vapour mixtures, and reactive systems are analyzed in Chapter 16. To increase the utility of the book, Chapters 17 and 18 dealing with compressible fluid flow and heat transfer respectively, have been added.

Many illustrative examples are solved and many problems are provided in each chapter to aid comprehension and to stimulate the interest of the students.

Throughout the text SI units have been used. Tables and charts given in the Appendix are also in SI units.

This book is based mainly on the lecture notes prepared for classes on the subject at IIT Kharagpur. I am grateful to the authors of the books that I used in preparing the notes, a list of which is given in the bibliography. I am thankful to my colleagues in the mechanical engineering department of IIT Kharagpur, for many stimulating discussions, and for encouragement. I am indebted to all those who have helped in the preparation of the book.

I would very much appreciate criticisms, suggestions for improvement, and detection of errors from my readers, which will be gratefully acknowledged.

P K Nag

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Acknowledgements

The author gratefully remembers the contributions made by Late Prof. Bindu Madhav Belgaumkar, the Head of the Department of Mechanical Engineering (1954–1970), at Indian Institute of Technology (IIT), Kharagpur, in building up his teaching career during his earlier formative years, by providing the right academic environment in the campus. He also humbly acknowledges the contribution of his teacher, Prof. R G. Mokadam for maintaining high standard of teaching Thermodynamics in the decades of 1950s and 1960s at IIT, Kharagpur, the first IIT which played the pioneering role in technical education in the country.

The author is indebted to his erstwhile colleagues at IIT for many positive interactions, some of whom were his teachers during his M.Tech course, viz., Prof. C N Lakshminarayan, Late Prof. H R Narayan, Prof. Jayanand H. Hiranandani, Prof. H V Rao, Prof. B G Ghosh, Prof. A K Roy, Late Prof. P K G Pannikar, Prof. Darshan Lal, Prof. N S Murthy and Prof. S K Som.

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CHAPTER 1

Introduction

Thermodynamics is the science of energy transfer and its effect on the physical properties of substances. It is based upon observations of common experience which have been formulated into thermodynamic laws. These laws govern the principles of energy conversion. The applications of the thermodynamic laws and principles are found in all fields of energy technology, notably in steam and nuclear power plants, internal combustion engines, gas turbines, air conditioning, refrigeration, gas dynamics, jet propulsion, compressors, chemical process plants, and direct energy conversion devices.

1.1 MACROSCOPIC VERSUS MICROSCOPIC VIEWPOINT

There are two points of view from which the behaviour of matter can be studied: the macroscopic and the microscopic. In the macroscopic approach, a certain quantity of matter is considered, without the events occurring at the molecular level being taken into account. From the microscopic point of view, matter is composed of myriads of molecules. If it is a gas, each molecule at a given instant has a certain position, velocity, and energy, and for each molecule these change very frequently as a result of collisions. The behaviour of the gas is described by summing up the behaviour of each molecule. Such a study is made in *microscopic or statistical thermodynamics*. *Macroscopic thermodynamics* is only concerned with the effects of the action of many molecules, and these effects can be perceived by human senses. For example, the macroscopic quantity, pressure, is the average rate of change of momentum due to all the molecular collisions made on a unit area. The effects of pressure can be felt. The macroscopic point of view is not concerned with the action of individual molecules, and the force on a given unit area can be measured by using, e.g. a pressure gauge. These macroscopic observations are completely independent of the assumptions regarding the nature of matter. All the results of classical or macroscopic thermodynamics can, however, be derived from the microscopic and statistical study of matter.

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1.2 THERMODYNAMIC SYSTEM AND CONTROL VOLUME

A thermodynamic *system* is defined as a quantity of matter or a region in space upon which attention is concentrated in the analysis of a problem. Everything external to the system is called the *surroundings* or the *environment*. The system is separated from the surroundings by the system boundary (Fig. 1.1). The boundary may be either *fixed* or *moving*. A system and its surroundings together comprise a *universe*.

There are three classes of systems: (a) closed system, (b) open system, and (c) isolated system. The *closed system* (Fig. 1.2) is a system of fixed mass. There is

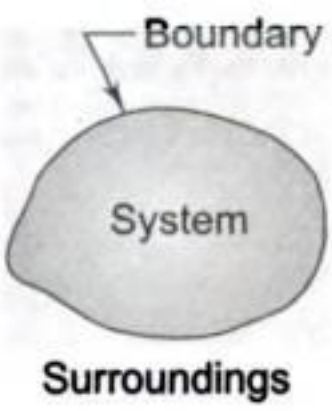

Fig. 1.1 *A Thermodynamic System*

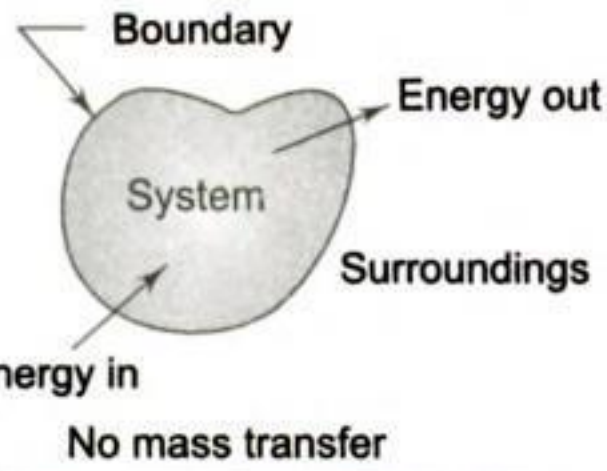

Fig. 1.2 *A Closed System*

no mass transfer across the system boundary. There may be energy transfer into or out of the system. A certain quantity of fluid in a cylinder bounded by a piston constitutes a closed system. The *open system* (Fig. 1.3) is one in which matter crosses the boundary of the system. There may be energy transfer also. Most of the engineering devices are generally open systems, e.g. an air compressor in which air enters at low pressure and leaves at high pressure and there are energy transfers across the system boundary. The *isolated system* (Fig. 1.4) is one in which there is no interaction between the system and the surroundings. It is of fixed mass and energy, and there is no mass or energy transfer across the system boundary.

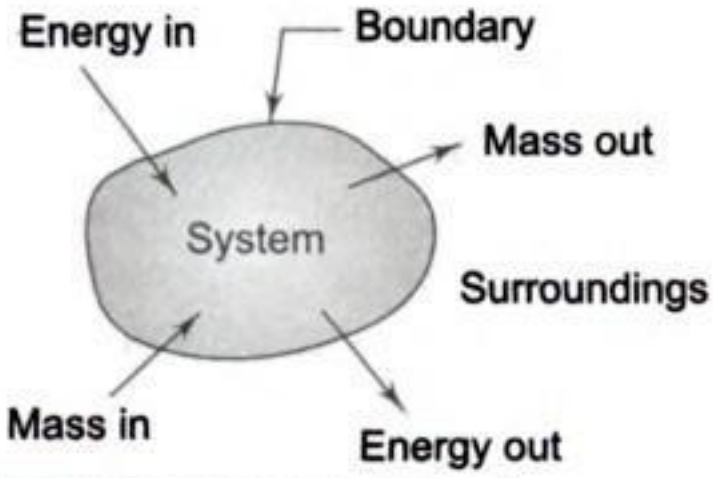

Fig. 1.3 *An Open System*

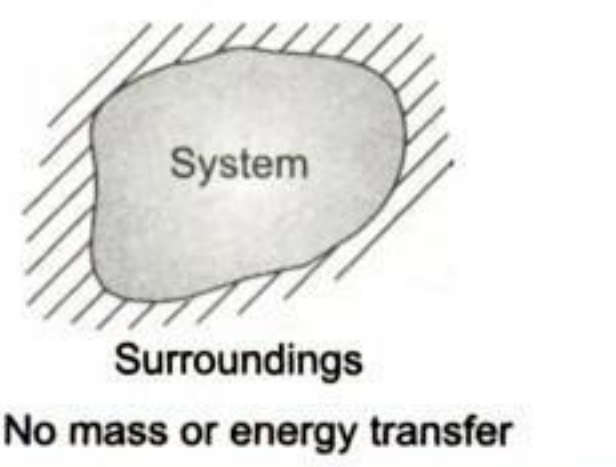

Fig. 1.4 *An Isolated System*

If a system is defined as a certain quantity of matter, then the system contains the same matter and there can be no transfer of mass across its boundary. However, if a system is defined as a region of space within a prescribed boundary, then matter can cross the system boundary. While the former is called a closed system, the latter is an open system.

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For thermodynamic analysis of an open system, such as an air compressor (Fig. 1.5), attention is focussed on a certain volume in space surrounding the compressor, known as the *control volume*, bounded by a surface called the *control surface*. Matter as well as energy crosses the control surface.

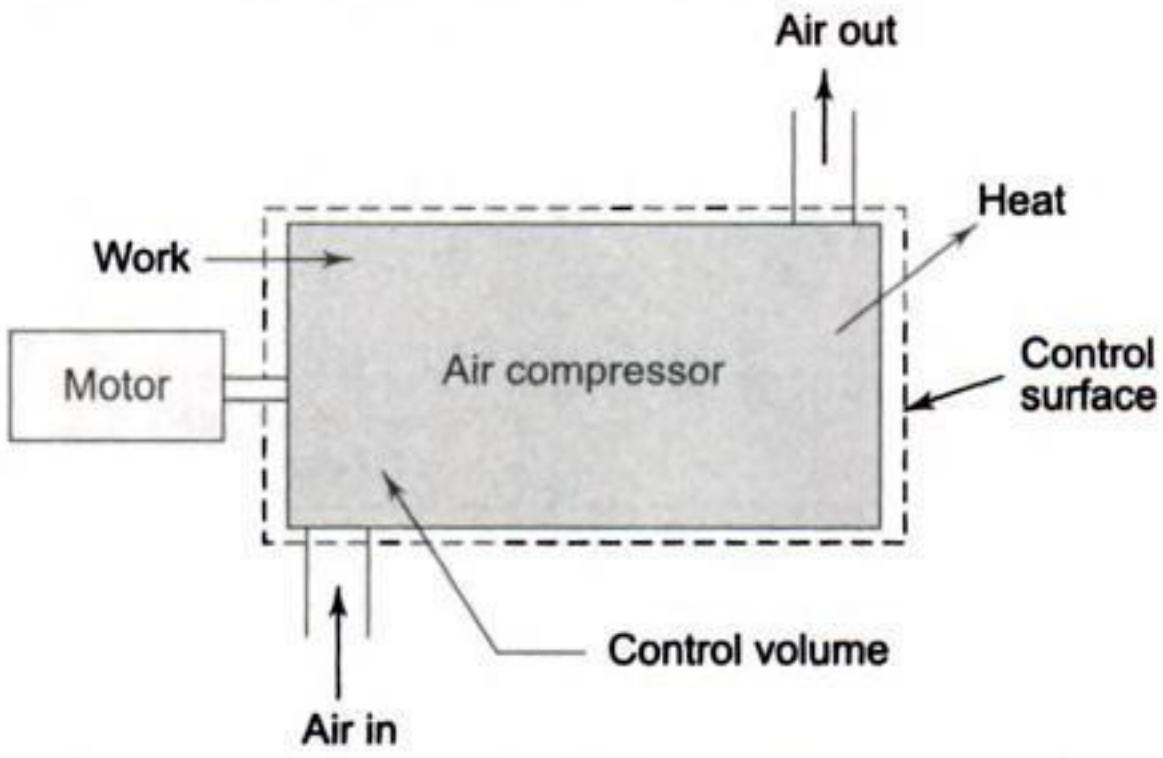

Fig. 1.5 *Control Volume and Control Surface*

A closed system is a system closed to matter flow, though its volume can change against a flexible boundary. When there is mater flow, then the system is considered to be a volume of fixed identity, the control volume. There is thus no difference between an open system and a control volume.

1.3 THERMODYNAMIC PROPERTIES, PROCESSES AND CYCLES

Every system has certain characteristics by which its physical condition may be described, e.g. volume, temperature, pressure, etc. Such characteristics are called *properties* of the system. These are all *macroscopic* in nature. When all the properties of a system have definite values, the system is said to exist at a definite *state*. Properties are the coordinates to describe the state of a system. They are the state variables of the system. Any operation in which one or more of the properties of a system changes is called a *change of state*. The succession of states passed through during a change of state is called the *path* of the change of state. When the path is completely specified, the change of state is called a *process*, e.g. a constant pressure process. A thermodynamic *cycle* is defined as a series of state changes such that the final state is identical with the initial state (Fig. 1.6).

Properties may be of two types. *Intensive properties* are independent of the mass in the system, e.g. pressure, temperature, etc. *Extensive properties* are related to mass, e.g. volume, energy, etc. If mass is increased, the values of the extensive properties also increase. Specific extensive properties, i.e. extensive properties per unit mass, are intensive properties, e.g. specific volume, specific energy, density, etc.

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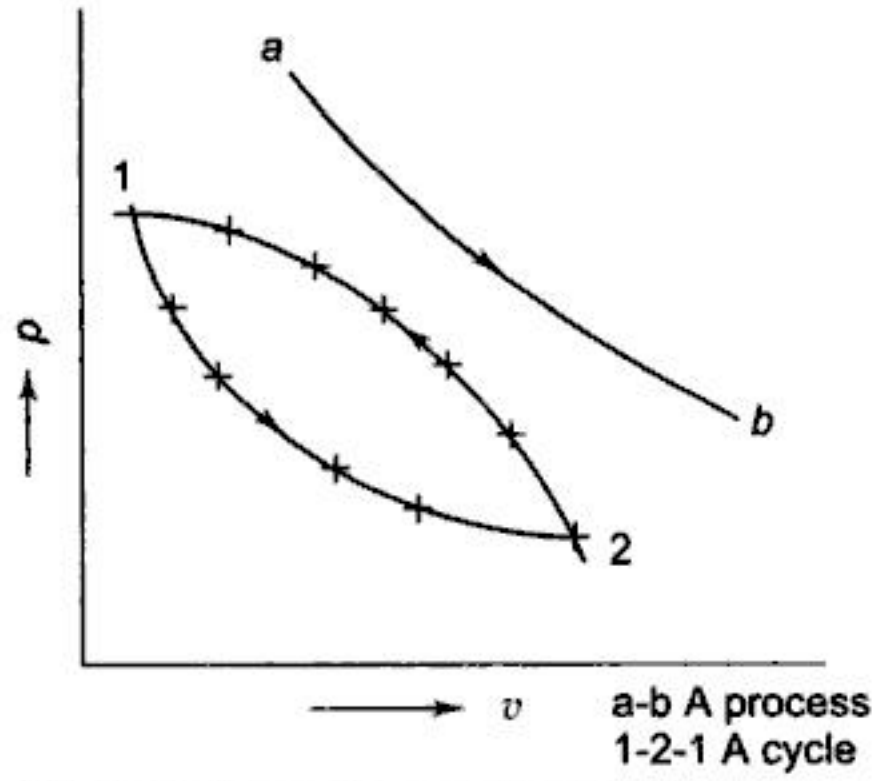

Fig. 1.6 *A Process and a Cycle*

1.4 HOMOGENEOUS AND HETEROGENEOUS SYSTEMS

A quantity of matter homogeneous throughout in chemical composition and physical structure is called a *phase*. Every substance can exist in any one of the three phases, viz. solid, liquid and gas. A system consisting of a single phase is called a *homogeneous system*, while a system consisting of more than one phase is known as a *heterogeneous system*.

1.5 THERMODYNAMIC EQUILIBRIUM

A system is said to exist in a state of *thermodynamic equilibrium* when no change in any macroscopic property is registered, if the system is isolated from its surroundings.

An isolated system always reaches in course of time a state of thermodynamic equilibrium and *can never depart from it spontaneously*.

Therefore, there can be no *spontaneous change in any macroscopic property* if the system exists in an equilibrium state. Thermodynamics studies mainly the properties of physical systems that are found in equilibrium states.

A system will be in a state of thermodynamic equilibrium, if the conditions for the following three types of equilibrium are satisfied:

(a) Mechanical equilibrium
(b) Chemical equilibrium
(c) Thermal equilibrium

In the absence of any unbalanced force within the system itself and also between the system and the surroundings, the system is said to be in a state of *mechanical equilibrium*. If an unbalanced force exists, either the system alone or both the system and the surroundings will undergo a change of state till mechanical equilibrium is attained.

If there is no chemical reaction or transfer of matter from one part of the system to another, such as diffusion or solution, the system is said to exist in a state of *chemical equilibrium*.

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When a system existing in mechanical and chemical equilibrium is separated from its surroundings by a diathermic wall (diathermic means 'which allows heat to flow') and if there is no spontaneous change in any property of the system, the system is said to exist in a state of *thermal equilibrium*. When this is not satisfied, the system will undergo a change of state till thermal equilibrium is restored.

When the conditions for any one of the three types of equilibrium are not satisfied, a system is said to be in a *nonequilibrium state*. If the nonequilibrium of the state is due to an unbalanced force in the interior of a system or between the system and the surroundings, the pressure varies from one part of the system to another. There is no single pressure that refers to the system as a whole. Similarly, if the nonequilibrium is because of the temperature of the system being different from that of its surroundings, there is a nonuniform temperature distribution set up within the system and there is no single temperature that stands for the system as a whole. It can thus be inferred that when the conditions for thermodynamic equilibrium are not satisfied, the states passed through by a system cannot be described by thermodynamic properties which represent the system as a whole.

Thermodynamic properties are the macroscopic coordinates defined for, and significant to, only thermodynamic equilibrium states. Both classical and statistical thermodynamics study mainly the equilibrium states of a system.

1.6 QUASI-STATIC PROCESS

Let us consider a system of gas contained in a cylinder (Fig. 1.7). The system initially is in an equilibrium state, represented by the properties p_1, v_1, t_1. The weight on the piston just balances the upward force exerted by the gas. If the weight is removed, there will be an unbalanced force between the system and the surroundings, and under gas pressure, the piston will move up till it hits the stops. The system again comes to an equilibrium state, being described by the properties p_2, v_2, t_2. But the intermediate states passed through by the system are nonequilibrium states which cannot be described by thermodynamic coordinates.

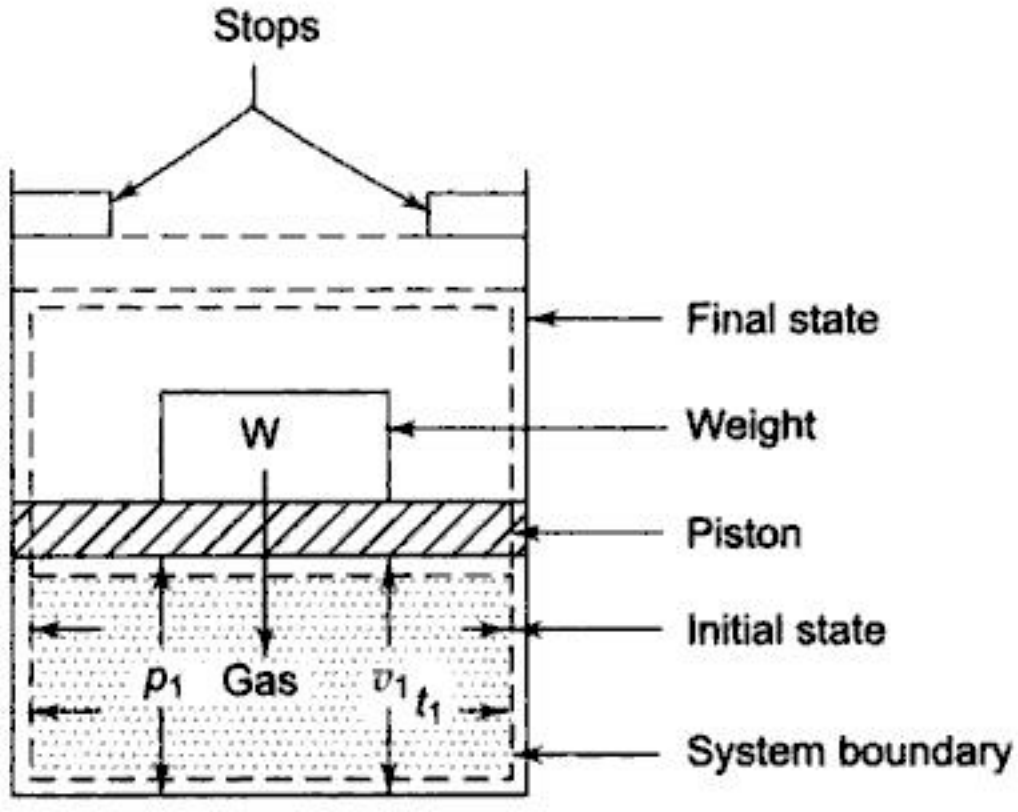

Fig. 1.7 *Transition between Two Equilibrium States by an Unbalanced Force*

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Figure 1.8 shows points 1 and 2 as the initial and final equilibrium states joined by a dotted line, which has got no meaning otherwise. Now if the single weight on the piston is made up of many very small pieces of weights (Fig. 1.9), and these weights are removed one by one very slowly from the top of the piston, at any instant of the upward travel of the piston, if the gas system is isolated, *the departure of the state of the system from the thermodynamic equilibrium state will be infinitesimally small.* So every state passed through by the system will be an

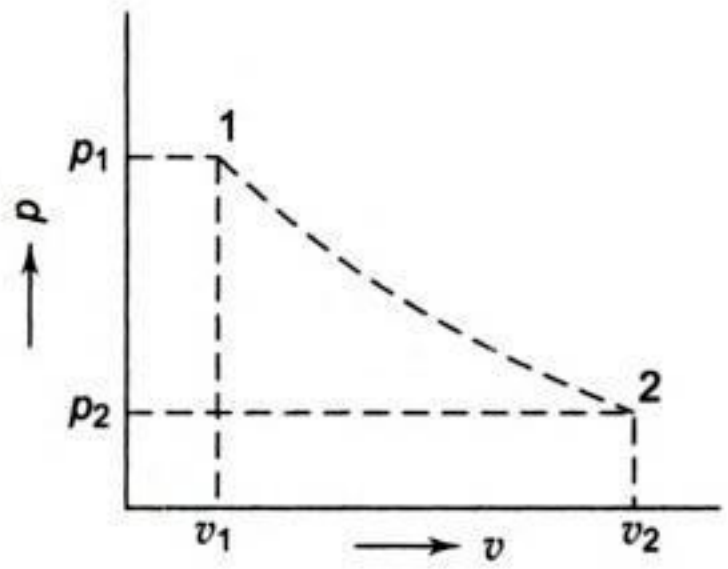

Fig. 1.8 *Plot Representing the Transition between Two Equilibrium States*

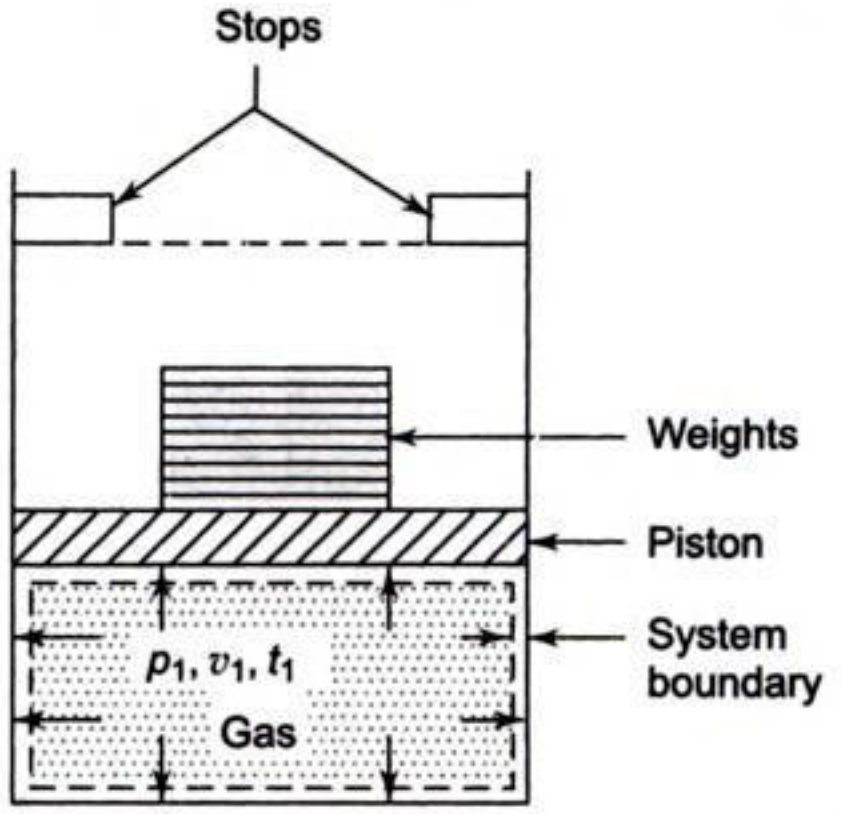

Fig. 1.9 *Infinitely Slow Transition of a System by Infinitesimal Force*

equilibrium state. Such a process, which is but a locus of all the equilibrium points passed through by the system, is known as a *quasi-static process* (Fig. 1.10), 'quasi' meaning 'almost'. *Infinite slowness is the characteristic feature of a quasi-static process.* A quasi-static process is thus a succession of equilibrium states. A quasi-static process is also called a *reversible process.*

1.7 PURE SUBSTANCE

A pure substance is defined as one that is homogeneous and invariable in chemical composition throughout its mass. The relative proportions of the chemical elements constituting the substance are also constant. Atmospheric air, steam-water mixture and combustion products of a fuel are regarded as pure substances. But the mixture of air and liquid air is not a pure substance, since the relative proportions of oxygen and nitrogen differ in the gas and liquid phases in equilibrium.

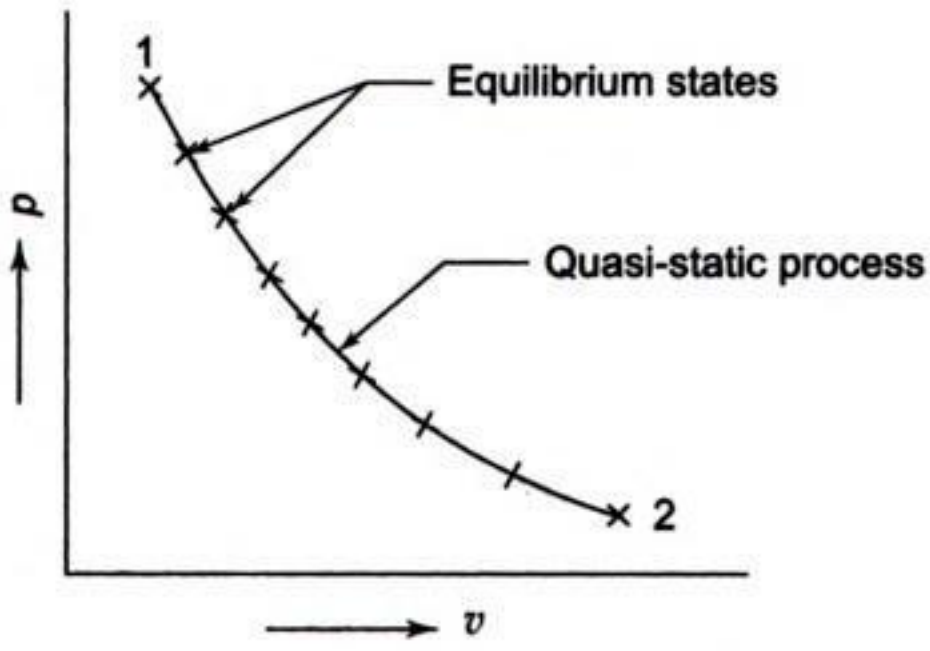

Fig. 1.10 *A Quasi-Static Process*

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The state of a pure substance of given mass can be fixed by specifying two independent intensive properties, provided the system is in equilibrium. This is known as the '*two-property rule*'. The state can thus be represented as a point on thermodynamic property diagrams. Once any two properties of a pure substance are known, other properties can be determined from the available thermodynamic relations.

1.8 CONCEPT OF CONTINUUM

From the macroscopic viewpoint, we are always concerned with volumes which are very large compared to molecular dimensions. Even a very small volume of a system is assumed to contain a large number of molecules so that statistical averaging is meaningful and a property value can be assigned to it. Disregarding the behaviour of individual molecules, matter is here treated as continuous. Let us consider the mass δm in a volume δV surrounding the point P (Fig. 1.11). The ratio $\delta m/\delta V$ is the average mass density of the system within the volume δV. We suppose that at first δV is rather large, and is subsequently shrunk about the point P. If we plot $\delta m/\delta V$ against δV, the average density tends to approach an asymptote as δV increases (Fig. 1.12). However, when δV becomes so small as to contain relatively few molecules, the average density fluctuates substantially with time as molecules pass into and out of the volume in random motion, and so it is impossible to speak of a definite value of $\delta m/\delta V$. The smallest volume which may be regarded as continuous is $\delta V'$. The density ρ of the system at a point is thus defined as

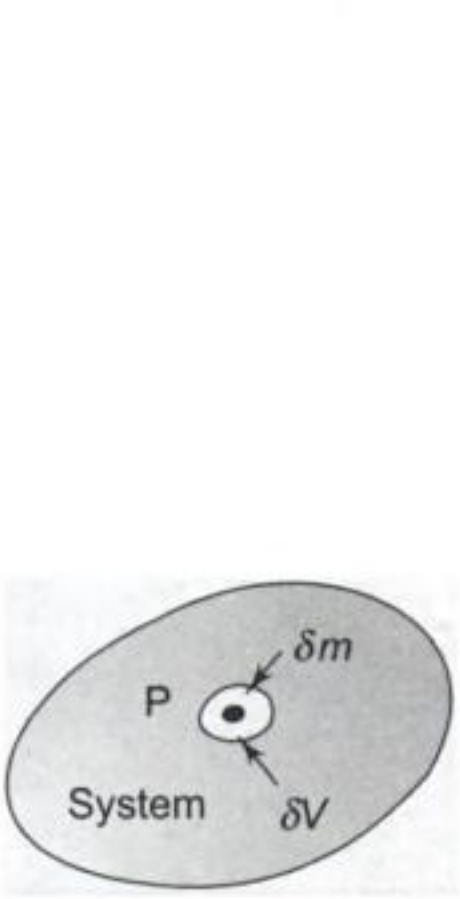

Fig. 1.11

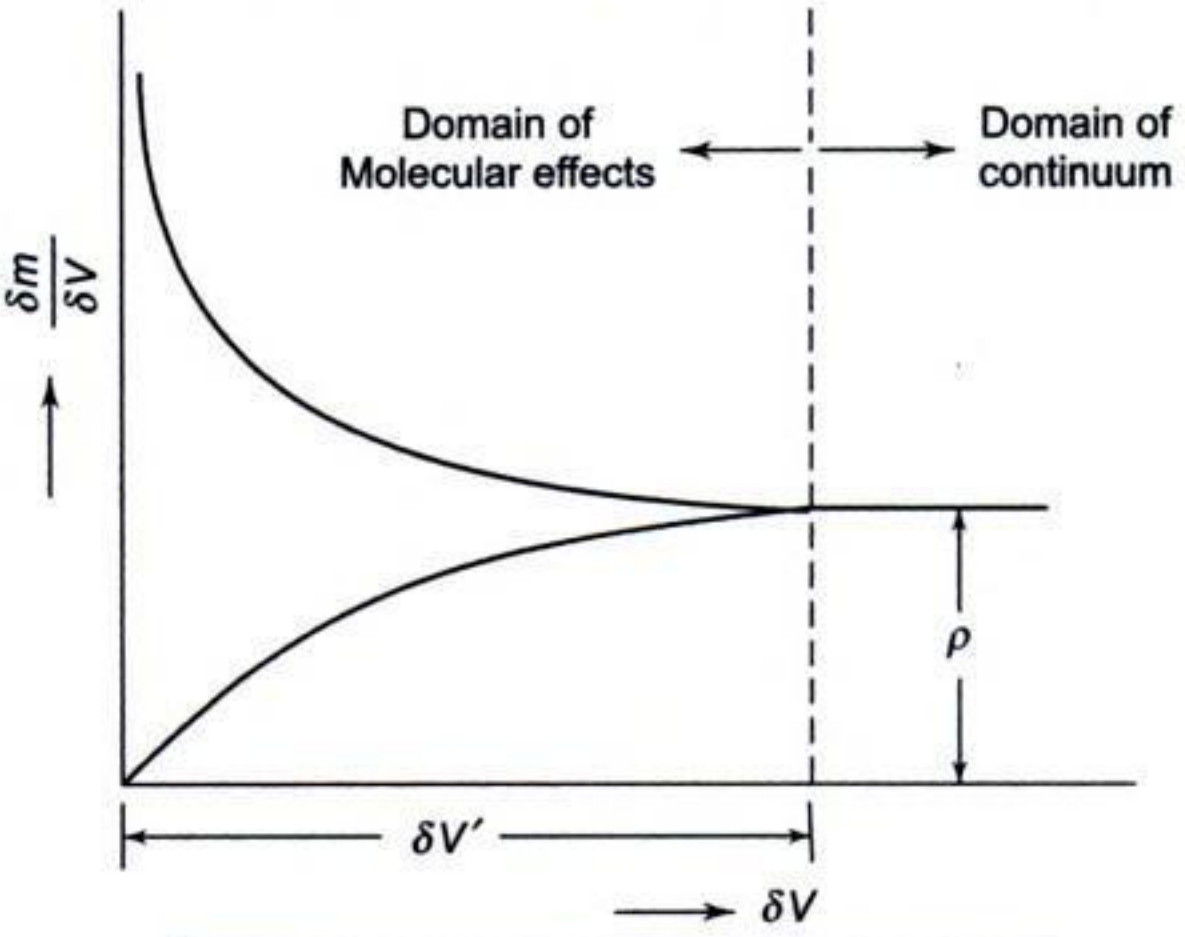

Fig.1.12 *Definition of the Macroscopic Property, Density*

$$\rho = \lim_{\delta V \to \delta V'} \frac{\delta m}{\delta V} \tag{1.1}$$

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Similarly, the fluid velocity at a point P is defined as the instantaneous velocity of the centre of gravity of the smallest continuous volume $\delta V'$.

The concept of continuum loses validity when the mean free path of the molecules approaches the order of magnitude of the dimensions of the vessel, as, for instance, in highly rarefied gases encountered in high vacuum technology, in rocket flights at high altitudes and in electron tubes. In most engineering applications, however, the assumption of a continuum is valid and convenient, and goes hand in hand with the macroscopic point of view.

1.9 THERMOSTATICS

The science of thermodynamics deals with systems existing in thermodynamic equilibrium states which are specified by properties. Infinitely slow quasi-static processes executed by systems are only meaningful in thermodynamic plots. The name 'thermodynamics' is thus said to be a misnomer, since it does not deal with the dynamics of heat, which is nonquasi-static. The name 'thermostatics' then seems to be more appropriate. However, most of the real processes are dynamic and nonquasi-static, although the initial and final states of the system might be in equilibrium. Such processes can be successfully dealt with by the subject. Hence, the term 'thermodynamics' is not inappropriate.

1.10 UNITS AND DIMENSIONS

In the present text, the SI (System International) system of units has been used. The basic units in this system are given in Table 1.1.

Table 1.1 *System: Basic Units*

Quantity	*Unit*	*Symbol*
Length (L)	Metre	m
Mass (M)	Kilogram	kg
Time (t)	Second	s
Amount of substance	Mole	mol
Temperature (T)	Kelvin	K
Electric current	Ampere	A
Luminous intensity	Candela	cd
Plane angle	Radian	rad
Solid angle	Steradian	sr

The dimensions of all other quantities are derived from these basic units which are given in Table 1.2.

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Table 1.2 *SI System: Derived Units*

Quantity	*Unit*	*Symbol*	*Alternative unit*	*In basic units*
Force (F)	Newton	N		$kg\ m/s^2$
Energy E)	Joule	J	Nm	$kg\ m^2/s^2$
Power	Watt	W	J/s	$kg\ m^2/s^3$
Pressure	Pascal	Pa	N/m^2	$kg/(ms^2)$
Frequency	Hertz	Hz		s^{-1}
Electric charge	Coulomb	C		As
Electric potential	Volt	V	W/A = J/C	$kg\ m^2/(s^3\ A)$
Capacitance	Farad	F	C/V	$s^4\ A^2/(kg\ m^2)$
Electrical resistance	Ohm	Ω	V/A	$kg\ m^2/(s^3\ A^2)$
Magnetic flux	Weber	Wb	Vs	$kg\ m^2/(s^2\ A)$
Magnetic flux density	Tesla	T	Wb/m^2	$kg/(s^2\ A)$
Inductance	Henry	H	Wb/A	$kg\ m^2/(s^2\ A^2)$

It is often convenient and desirable to use multiples of various units, the standard list of which is given in Table 1.3.

Table 1.3 *SI System: Standard Multipliers*

Factor	*Prefix*	*Factor*	*Prefix*
10^{12}	tera, T	10^{-3}	milli, m
10^{9}	giga, G	10^{-6}	micro, μ
10^{6}	mega, M	10^{-9}	nano, n
10^{3}	kilo, k	10^{-12}	pico, p

1.10.1 Force

The force acting on a body is defined by Newton's second law of motion. The unit of force is the newton (N). A force of one newton produces an acceleration of 1 ms^{-2} when applied to a mass of 1 kg.

$$1\ N = 1\ kg\ m/s^2$$

The weight of a body (W) is the force with which the body is attracted to the centre of the earth. It is the product of its mass (m) and the local gravitational acceleration (g), i.e.

$$W = mg$$

The value of g at sea level is 9.80665 m/s^2. The mass of a substance remains constant with elevation, but its weight varies with elevation.

1.10.2 Pressure

Pressure is the normal force exerted by a system against unit area of the bounding surface. If δA is a small area and $\delta A'$ is the smallest area from continuum consideration, and δF_n is the component of force normal to δA (Fig. 1.13), the pressure p at a point on the wall is defined as

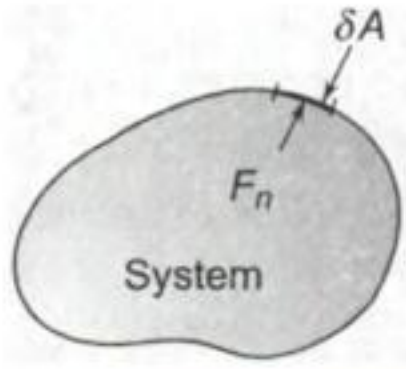

Fig. 1.13 *Definition of Pressure*

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$$p = \lim_{\delta A \to \delta A'} \frac{\delta F_n}{\delta A}$$

The pressure p at a point in a fluid in equilibrium is the same in all directions.

The unit for pressure in the SI system is the *pascal* (Pa), which is the force of one newton acting on an area of 1 m^2.

$$1 \text{ Pa} = 1 \text{ N/m}^2$$

The unit of pascal is very small. Very often kilo-pascal (kPa) or mega-pascal (MPa) is used.

Two other units, not within the SI system of units, continue to be widely used. These are the *bar*, where

$$1 \text{ bar} = 10^5 \text{ Pa} = 100 \text{ kPa} = 0.1 \text{ MPa}$$

and the standard atmosphere, where

$$1 \text{ atm} = 101.325 \text{ kPa} \qquad = 1.01325 \text{ bar}$$

Most instruments indicate pressure relative to the atmospheric pressure, whereas the pressure of a system is its pressure above zero, or relative to a perfect vacuum (Fig. 1.14). The pressure relative to the atmosphere is called *gauge pressure*. The pressure relative to a perfect vacuum is called *absolute pressure*.

Absolute pressure = Gauge pressure + Atmospheric pressure

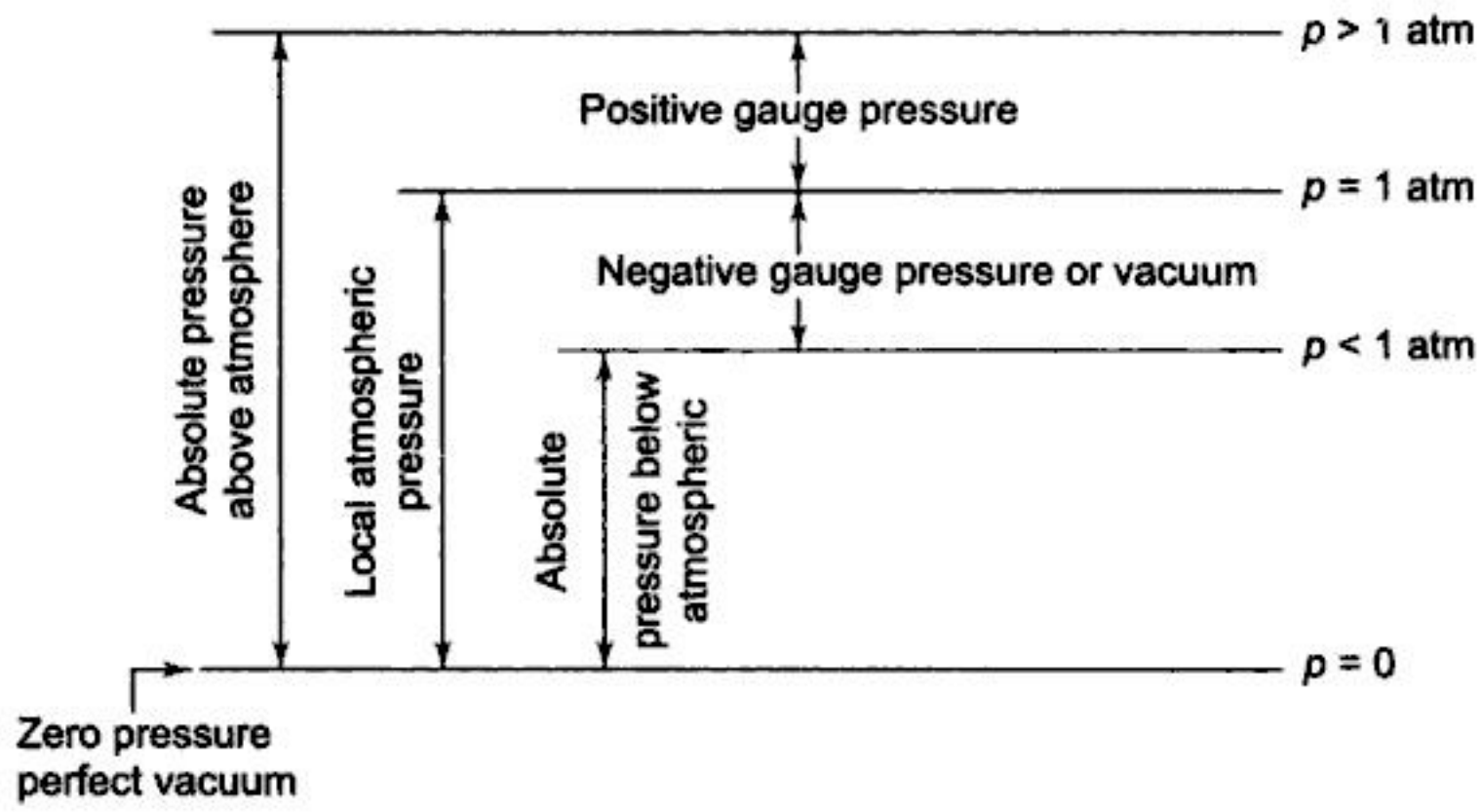

Fig. 1.14 *Relationship between Pressures*

When the pressure in a system is less than atmospheric pressure, the gauge pressure becomes negative, but is frequently designated by a positive number and called *vacuum*. For example, 16 cm vacuum will be

$$\frac{76 - 16}{76} \times 1.013 = 0.80 \text{ bar}$$

Thus,

$$p_{abs} = p_{atm} - p_{vac}$$

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Figure 1.15 shows a few pressure measuring devices. Figure (a) shows the Bourdon gauge which measures the difference between the system pressure inside the tube and atmospheric pressure. It relies on the deformation of a bent hollow tube of suitable material which, when subjected to the pressure to be measured on the inside (and atmospheric pressure on the outside), tends to unbend. This moves a pointer through a suitable gear-and lever mechanism against a calibrated scale. Figure (b) shows an open U-tube indicating gauge pressure, and Fig. (c) shows an open U-tube indicating vacuum. Figure (d) shows a closed U-tube indicating absolute pressure. If p is atmospheric pressure, this is a *barometer*. These are called U-tube manometers.

Fig. 1.15 *Pressure Gauges (a) Bourdon Gauge (b) Open U-tube Indicating Gauge Pressure (c) Open U-tube Indicating Vacuum (d) Closed U-tube Indicating Absolute Pressure*

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If Z is the difference in the heights of the fluid columns in the two limbs of the U-tube (Fig. (b) and Fig. (c)], ρ the density of the fluid and g the acceleration due to gravity, then from the elementary principle of hydrostatics, the gauge pressure p_g is given by

$$p_g = Z\rho g \left[\text{m} \cdot \frac{\text{kg}}{\text{m}^3} \frac{\text{m}}{\text{s}^2} \right] = Z\rho g \text{ N/m}^2$$

If the fluid is mercury having $\rho = 13{,}616$ kg/m^3, one metre head of mercury column is equivalent to a pressure of 1.3366 bar, as shown below

$$1 \text{ m Hg} = Z\,\rho\,g = 1 \times 13616 \times 9.81$$
$$= 1.3366 \times 10^5 \text{ N/m}^2 = 1.3366 \text{ bar}$$

The manometer is a sensitive, accurate and simple device, but it is limited to fairly small pressure differentials and, because of the inertia and friction of the liquid, is not suitable for fluctuating pressures, unless the rate of pressure change is small. A diaphragm-type pressure transducer along with a cathode ray oscilloscope can be used to measure rapidly fluctuating pressures. The unit of 1 mm Hg pressure is called *torr*, so that

$$1 \text{ mm Hg} = 1 \text{ torr} = 133 \text{ Pa}$$

Figure 1.16 shows a typical U-tube manometer, one end of which is connected to the vessel the pressure of which is to be measured and the other end is open to atmosphere exerting pressure p_{atm}. The manometric fluid may be mercury, water, alcohol, oil, etc. The pressure along the horizontal line AB is the same in both the limbs of the manometer, so that

$$p + \rho_1 g z_1 = p_{atm} + \rho_2 g z_2$$

If the vessel contains a gas of density ρ_1 and since the density of the manometric fluid, (say Hg), $\rho_2 >> \rho_1$,

$$p - p_{atm} = \rho_2 g z_2 = p_{gauge}$$

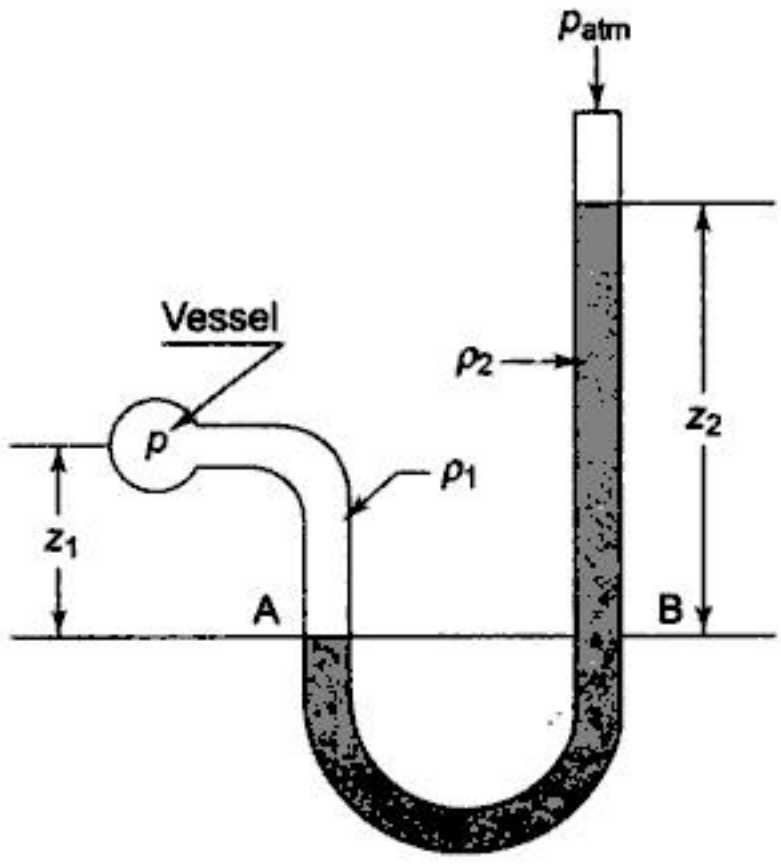

Fig. 1.16 *Pressure Measurement by a Manometer*

1.10.3 Specific Volume and Density

Volume (V) is the space occupied by a substance and is measured in m^3. The specific volume (v) of a substance is defined as the volume per unit mass and is measured in m^3/kg. From continuum consideration the specific volume at a point is defined as

$$v = \lim_{\delta V \to \delta V'} \frac{\delta V}{\delta m}$$

where dV' is the smallest volume for which the system can be considered a continuum.

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Density (ρ) is the mass per unit volume of a substance, which has been discussed earlier, and is given in kg/m^3.

$$\rho = \frac{m}{v}$$

In addition to m^3, another commonly used unit of volume is the litre (l).

$$1\ \text{l} = 10^{-3}\ \text{m}^3$$

The specific volume or density may be given either on the basis of mass or in respect of mole. A *mole* of a substance has a mass numerically equally to the molecular weight of the substance. One g mol of oxygen has a mass of 32 g and 1 kg mol (or kmol) of nitrogen has a mass of 28 kg. The symbol $\bar{v}$ is used for molar specific volume (m^3/kmol).

1.10.4 Energy

Energy is the capacity to exert a force through a distance, and manifests itself in various forms. Engineering processes involve the conversion of energy from one form to another, the transfer of energy from place to place, and the storage of energy in various forms, utilizing a working substance.

The unit of energy in the SI system is Nm or J (joule). The energy per unit mass is the specific energy, the unit of which is J/kg.

1.10.5 Power

The rate of energy transfer or storage is called power. The unit of power is watt (W), kilowatt (kW) or megawatt (MW).

$$1\ \text{W} = 1\ \text{J/s} = 1\ \text{Nm/s}$$

$$1\ \text{kW} = 1000\ \text{W}$$

Solved Examples

Example 1.1 The pressure of gas in a pipe line is measured with a mercury manometer having one limb open to the atmosphere (Fig. 1.17). If the difference in the height of mercury in the two limbs is 562 mm, calculate the gas pressure. The barometer reads 761 mm Hg, the acceleration due to gravity is 9.79 m/s^2, and the density of mercury is 13,640 kg/m^3.

Solution At the plane AB, we have

$$p = p_0 + \rho\, g\, z$$

Now

$$p_0 = \rho\, g\, z_0$$

where z_0 is the barometric height, ρ the density of mercury and p_0 the atmospheric pressure.

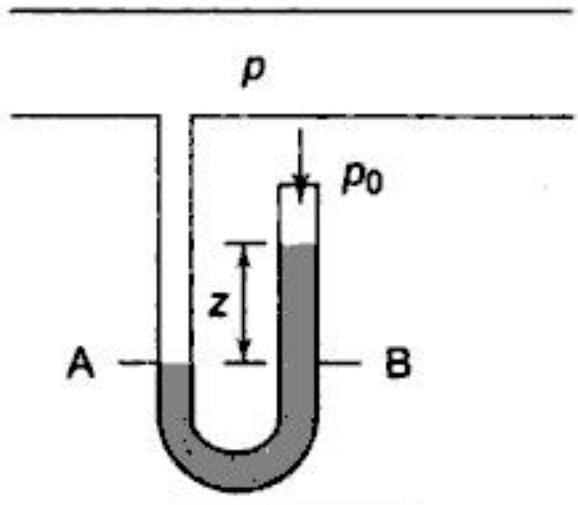

Fig. 1.17

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Therefore

$$p = \rho g (z + z_0)$$
$$= 13{,}640 \text{ kg/m}^3 \times 9.79 \text{ m/s}^2 (0.562 + 0.761) \text{ m}$$
$$= 177 \times 10^3 \text{ N/m}^2 = 177 \text{ kPa} = 1.77 \text{ bar} = \mathbf{1.746 \text{ atm}}$$

Example 1.2 A turbine is supplied with steam at a gauge pressure of 1.4 MPa. After expansion in the turbine the steam flows into a condenser which is maintained at a vacuum of 710 mm Hg. The barometric pressure is 772 mm Hg. Express the inlet and exhaust steam pressures in pascals (absolute). Take the density of mercury as 13.6×10^3 kg/m^3.

Solution The atmospheric pressure p_0

$$= \rho g z_0 = 13.6 \times 10^3 \text{ kg/m}^3 \times 9.81 \text{ m/s}^2 \times 0.772 \text{ m}$$
$$= 1.03 \times 10^5 \text{ Pa}$$

Inlet steam pressure

$$= [(1.4 \times 10^6) + (1.03 \times 10^5)] \text{ Pa}$$
$$= 15.03 \times 10^5 \text{ Pa} = \mathbf{1.503 \text{ MPa}}$$

Condenser pressure

$$= (0.772 - 0.710) \text{ m} \times 9.81 \text{ m/s}^2 \times 13.6 \times 10^3 \text{ kg/m}^3$$
$$= 0.827 \times 10^4 \text{ Pa} = \mathbf{8.27 \text{ kPa}}$$

Example 1.3 Convert the following readings of pressure to kPa, assuming that the barometer reads 760 mm of Hg.

(a) 40 cm Hg vacuum
(b) 90 cm Hg gauge
(c) 1.2 m of H_2O gauge

Solution (a)

$$p_{\text{vacuum}} = h\rho_g = (40 \times 10^{-2}) \times (13.6 \times 10^3) \times 9.8$$
$$= 53.31 \times 10^3 \text{ N/m}^2 = \mathbf{53.31 \text{ kPa}}$$

$$p_{\text{absolute}} = p_{\text{atm}} - p_{\text{vac}}$$
$$= (760 - 400) \times 9.8 \times 13.6 \times 10^3$$
$$= 48 \times 10^3 \text{ N/m}^2 = \mathbf{48 \text{ kPa}}$$

Also, $p_{\text{abs}} = 101.325 - 53.31 = 48.015$ kPa

(b) $p_{\text{gauge}} = h \rho g$
$$= (90 \times 10^{-2}) \times (13.6 \times 10^3) \times 9.8$$
$$= 120 \times 10^3 \text{ N/m}^2 = \mathbf{120 \text{ kPa}}$$

Since $p_{\text{atm}} = 760$ mm Hg $= 101.325$ kPa

$$p_{\text{abs}} = p_{\text{gauge}} + p_{\text{atm}}$$
$$= 120 + 101.325 = \mathbf{221.325 \text{ kPa}}$$

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(c) $$p_{\text{gauge}} = h\,\rho\,g = 1.2\ \text{m} \times 1000\ \frac{\text{kg}}{\text{m}^3} \times 9.81\ \frac{\text{m}}{\text{s}^2}$$

$$= 11.772\ \text{kPa}$$

$$\therefore \quad p_{\text{abs}} = 11.772 + 101.325 = \mathbf{113.097\ kPa}$$

Example 1.4 The pressure and specific volume of the atmosphere are related according to the equation $pv^{1.4} = 2.5 \times 10^5$, where p is in N/m^2 and v in m^3/kg. What depth of atmosphere is required to produce a pressure of 1.033 bar at the earth's surface? Assume $g = 9.81\ m/s^2$.

Solution Let H be the depth of the atmosphere required to produce the pressure of 1.033 bar at the earth's surface. Considering an element of length dh (Fig. 1.18), by force balance:

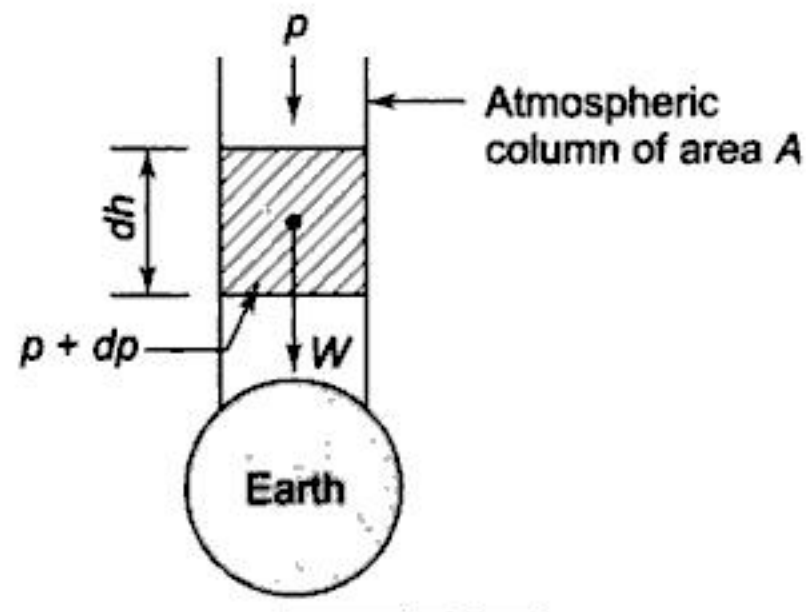

Fig. 1.18

$$A\,(p + dp) = mg + pA$$

$$pA + Adp = \rho A dh g + pA$$

or, $$dh = \frac{1}{g}\,v dp$$

or, $$\int_0^H dh = \int_0^{1.033\times10^5} \frac{1}{g}\left(\frac{2.5\times10^5}{p}\right)^{\frac{1}{1.4}} dp$$

$$H = \frac{1}{g}\,(2.5\times10^5)^{0.714} \int_0^{1.033\times10^5} p^{-0.714}\,dp$$

$$= 731.274\ \frac{\left(1.033\times10^5\right)^{0.286}}{0.286} = 69{,}462\ \text{m} = \mathbf{69.462\ km}$$

Review Questions

1.1 What do you understand by macroscopic and microscopic viewpoints?
1.2 Is thermodynamics a misnomer for the subject?
1.3 How does the subject of thermodynamics differ from the concept of heat transfer?
1.4 What is the scope of classical thermodynamics?
1.5 What is a thermodynamic system?
1.6 What is the difference between a closed system and an open system?
1.7 An open system defined for a fixed region and a control volume are synonymous. Explain.
1.8 Define an isolated sysetm.
1.9 Distinguish between the terms 'change of state', 'path', and 'process'.
1.10 What is a thermodynamic cycle?

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1.11 What are intensive and extensive properties?
1.12 What do you mean by homogeneous and heterogeneous systems?
1.13 Explain what you understand by thermodynamic equilibrium.
1.14 Explain mechanical, chemical and thermal equilibrium.
1.15 What is a quasi-static process? What is its characteristic feature?
1.16 What is the concept of continuum? How will you define density and pressure using this concept?
1.17 What is vacuum? How can it be measured?
1.18 What is a pressure transducer?

Problems

1.1 A pump discharges a liquid into a drum at the rate of 0.032 m^3/s. The drum, 1.50 m in diameter and 4.20 m in length, can hold 3000 kg of the liquid. Find the density of the liquid and the mass flow rate of the liquid handled by the pump. Ans. 12.934 kg/s

1.2 The acceleration of gravity is given as a function of elevation above sea level by

$$g = 980.6 - 3.086 \times 10^{-6} H$$

where g is in cm/s^2, and H is in cm. If an aeroplane weighs 90,000 N at sea level, what is the gravity force upon it at 10,000 m elevation? What is the percentage difference from the sea-level weight? Ans. 89,716.4 N, 0.315%

1.3 Prove that the weight of a body at an elevation H above sea-level is given by

$$W = \frac{mg}{g_0}\left(\frac{d}{d+2H}\right)^2$$

where d is the diameter of the earth.

1.4 The first artificial earth satellite is reported to have encircled the earth at a speed of 28,840 km/h and its maximum height above the earth's surface was stated to be 916 km. Taking the mean diameter of the earth to be 12,680 km, and assuming the orbit to be circular, evaluate the value of the gravitational acceleration at this height.

The mass of the satellite is reported to have been 86 kg at sea-level. Estimate the gravitational force acting on the satellite at the operational altitude.

Ans. 8.9 m/s^2; 765 N

1.5 Convert the following readings of pressure to kPa, assuming that the barometer reads 760 mm Hg:

(a) 90 cm Hg gauge, (b) 40 cm Hg vacuum, (c) 1.2 m H_2O gauge, (d) 3.1 bar.

1.6 A 30 m high vertical column of a fluid of density 1878 kg/m^3 exists in a place where $g = 9.65$ m/s^2. What is the pressure at the base of the column.

Ans. 544 kPa

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1.7 Assume that the pressure p and the specific volume v of the atmosphere are related according to the equation $pv^{1.4} = 2.3 \times 10^5$, where p is in N/m^2 abs and v is in m^3/kg. The acceleration due to gravity is constant at 9.81 m/s^2. What is the depth of atmosphere necessary to produce a pressure of 1.0132 bar at the earth's surface? Consider the atmosphere as a fluid column.

Ans. 64.8 km

1.8 The pressure of steam flowing in a pipeline is measured with a mercury manometer, shown in Fig. 1.19. Some steam condenses into water. Estimate the steam pressure in kPa. Take the density of mercury as 13.6×10^3 kg/m^3, density of water as 10^3 kg/m^3, the barometer reading as 76.1 cm Hg, and g as 9.806 m/s^2.

1.9 A vacuum gauge mounted on a condenser reads 0.66 m Hg. What is the absolute pressure in the condenser in kPa when the atmospheric pressure is 101.3 kPa?

Ans. 13.3 kPa

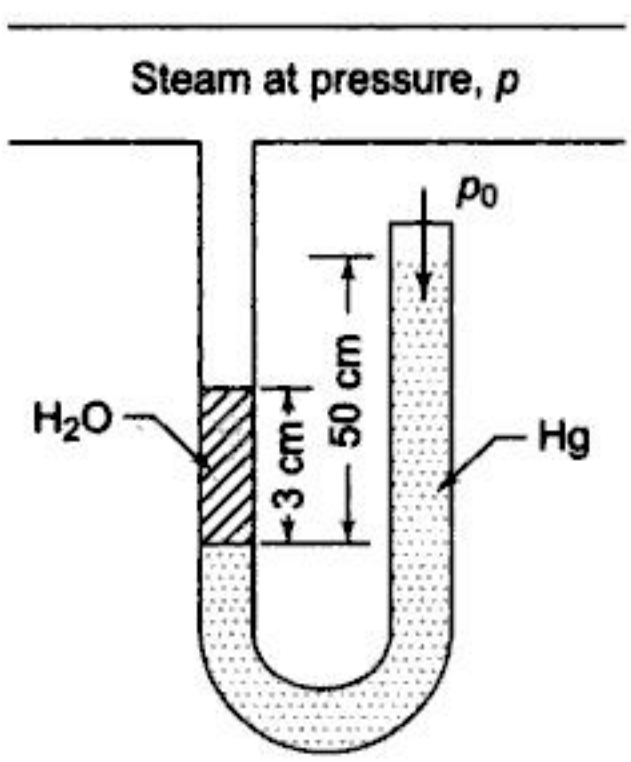

Fig. 1.19

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CHAPTER 2

Temperature

2.1 ZEROTH LAW OF THERMODYNAMICS

The property which distinguishes thermodynamics from other sciences is temperature. One might say that temperature bears as important a relation to thermodynamics as force does to statics or velocity does to dynamics. Temperature is associated with the ability to distinguish hot from cold. When two bodies at different temperatures are brought into contact, after some time they attain a common temperature and are then said to exist in thermal equilibrium.

When a body A is in thermal equilibrium with a body B, and also separately with a body C, then B and C will be in thermal equilibrium with each other.

This is known as the *zeroth law of thermodynamics*. It is the basis of temperature measurement.

In order to obtain a quantitative measure of temperature, a reference body is used, and a certain physical characteristic of this body which changes with temperature is selected. The changes in the selected characteristic may be taken as an indication of change in temperature. The selected characteristic is called the *thermometric property*, and the reference body which is used in the determination of temperature is called the *thermometer*. A very common thermometer consists of a small amount of mercury in an evacuated capillary tube. In this case the extension of the mercury in the tube is used as the thermometric property.

There are five different kinds of thermometer, each with its own thermometric property, as shown in Table 2.1.

2.2 MEASUREMENT OF TEMPERATURE—THE REFERENCE POINTS

The temperature of a system is a property that determines whether or not a system is in thermal equilibrium with other systems. If a body is at, say, 70°C, it will be 70°C, whether measured by a mercury-in-glass thermometer, resistance thermometer or constant volume gas thermometer. If X is the thermometric property, let us arbitrarily choose for the temperature common to the thermometer and *to all systems in thermal equilibrium with it* the following *linear* function of X:

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Table 2.1 *Thermometers and Thermometric Properties*

Thermometer	*Thermometric property*	*Symbol*
1. Constant volume gas thermometer	Pressure	p
2. Constant pressure gas thermometer	Volume	V
3. Electrical resistance thermometer	Resistance	R
4. Thermocouple	Thermal e.m.f.	ε
5. Mercury-in-glass thermometer	Length	L

$\theta(X) = aX$, where a is an arbitrary constant.

If X_1 corresponds to $\theta(X_1)$, then X_2 will correspond to

$$\frac{\theta(X_1)}{X_1} \cdot X_2$$

that is
$$\theta(X_2) = \frac{\theta(X_1)}{X_1} \cdot X_2 \tag{2.1}$$

Two temperatures on the linear X scale are to each other as the ratio of the corresponding X's.

2.2.1 Method Used Before 1954

The thermometer is first placed in contact with the system whose temperature $\theta(X)$ is to be measured, and then in contact with an arbitrarily chosen standard system in an easily reproducible state where the temperature is $\theta(X_1)$. Thus

$$\frac{\theta(X_1)}{\theta(X)} = \frac{X_1}{X} \tag{2.2}$$

Then the thermometer at the temperature $\theta(X)$ is placed in contact with another arbitrarily chosen standard system in another easily reproducible state where the temperature is $\theta(X_2)$. It gives

$$\frac{\theta(X_2)}{\theta(X)} = \frac{X_2}{X} \tag{2.3}$$

From Eqs. (2.2) and (2.3)

$$\frac{\theta(X_1) - \theta(X_2)}{\theta(X)} = \frac{X_1 - X_2}{X}$$

or
$$\theta(X) = \frac{\theta(X_1) - \theta(X_2)}{X_1 - X_2} \cdot X \tag{2.4}$$

If we assign an arbitrary number of degrees to the temperature interval $\theta(X_1) - \theta(X_2)$, then $\theta(X)$ can be calculated from the measurements of X, X_1 and X_2.

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An easily reproducible state of an arbitrarily chosen standard system is called a fixed point. Before 1954, there were two fixed points: (a) *the ice point*, the temperature at which pure ice coexisted in equilibrium with air-saturated water at one atmosphere pressure, and (b) *the steam point*, the temperature of equilibrium between pure water and pure steam at one atmosphere pressure. The temperature interval, $\theta(X_1) - \theta(X_2)$, between these two fixed points was chosen to be 100 degrees.

The use of two fixed points was found unsatisfactory and later abandoned, because of (a) the difficulty of achieving equilibrium between pure ice and air-saturated water (since when ice melts, it surrounds itself only with pure water and prevents intimate contact with air-saturated water), and (b) extreme sensitiveness of the steam point to the change in pressure.

2.2.2 Method in Use After 1954

Since 1954 only one fixed point has been in use, viz. *the triple point of water*, the state at which ice, liquid water and water vapour coexist in equilibrium. The temperature at which this state exists is arbitrarily assigned the value of 273.16 degrees Kelvin, or 273.16 K (the reason for using Kelvin's name will be explained later). Designating the triple point of water by θ_t, and with X_t being the value of the thermometric property when the body, whose temperature θ is to be measured, is placed in contact with water at its triple point, it follows that

$$\theta_t = aX_t$$

$$\therefore \quad a = \frac{\theta_t}{X_t} = \frac{273.16}{X_t}$$

Therefore

$$\theta = aX = \frac{273.16}{X_t} \cdot X$$

or

$$\theta = 273.16 \frac{X}{X_t} \tag{2.5}$$

The temperature of the triple point of water, which is an easily reproducible state, is now the *standard fixed point of thermometry.*

2.3 COMPARISON OF THERMOMETERS

Applying the above principle to the five thermometers listed in Table 2.1, the temperatures are given as follows:

(a) Constant volume gas thermometer $\quad \theta(P) = 273.16 \frac{p}{p_t}$

(b) Constant pressure gas thermometer $\quad \theta(V) = 273.16 \frac{V}{V_t}$

(c) Electric resistance thermometer $\quad \theta(R) = 273.16 \frac{R}{R_t}$

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(d) Thermocouple $\theta(\varepsilon) = 273.16 \dfrac{\varepsilon}{\varepsilon_t}$

(e) Liquid-in-glass thermometer $\theta(L) = 273.16 \dfrac{L}{L_t}$

If the temperature of a given system is measured simultaneously with each of the five thermometers, it is found that there is considerable difference among the readings. The smallest variation is, however, observed among different gas thermometers. That is why a gas is chosen as the standard thermometric substance.

2.4 IDEAL GAS

It has been established from experimental observations that the $p - v - T$ behaviour of gases at a low pressure is closely given by the following relation

$$p\bar{v} = \bar{R}T \tag{2.6}$$

where $\bar{R}$ is the universal gas constant, 8.3143 J/mol K and $\bar{v}$ is the molar specific volume, m^3/gmol. (see Sec. 10.3.). Dividing Eq. (2.6) by the molecular weight μ,

$$pv = RT \tag{2.7}$$

where v is specific volume, in m^3/kg, and R is the characteristic gas constant. Substituting $R = \bar{R}/\mu$ J/kg K, we get in terms of the total volume V of gas,

$$PV = n\bar{R}T$$

$$PV = mRT \tag{2.8}$$

where n is the number of moles and m is the mass of the gas. Equation (2.8) can be written for two states of the gas,

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2} \tag{2.9}$$

Equation (2.6), (2.7) or (2.8) is called the *ideal gas equation of state*. At very low pressure or density, all gases and vapours approach ideal gas behaviour.

2.5 GAS THERMOMETERS

A schematic diagram of a constant volume gas thermometer is given in Fig. 2.1. A small amount of gas is enclosed in bulb B which is in communication via the capillary tube C with one limb of the mercury manometer M. The other limb of the mercury manometer is open to the atmosphere and can be moved vertically to adjust the mercury levels so that the mercury just touches lip L of the capillary. The pressure in the bulb is used as a thermometric property and is given by

$$p = p_0 + \rho_M Zg$$

where p_0 is the atmospheric pressure ρ_M is the density of mercury.

When the bulb is brought in contact with the system whose temperature is to be measured, the bulb, in course of time, comes in thermal equilibrium with the system.

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The gas in the bulb expands, on being heated, pushing the mercury downward. The flexible limb of the manometer is then adjusted so that the mercury again touches the lip *L*. The difference in mercury level *Z* is recorded and the pressure *p* of the gas in the bulb is estimated. Since the volume of the trapped gas is constant, from the ideal gas equation,

$$\Delta T = \frac{V}{R}\Delta p \qquad (2.10)$$

i.e. the temperature increase is proportional to the pressure increase.

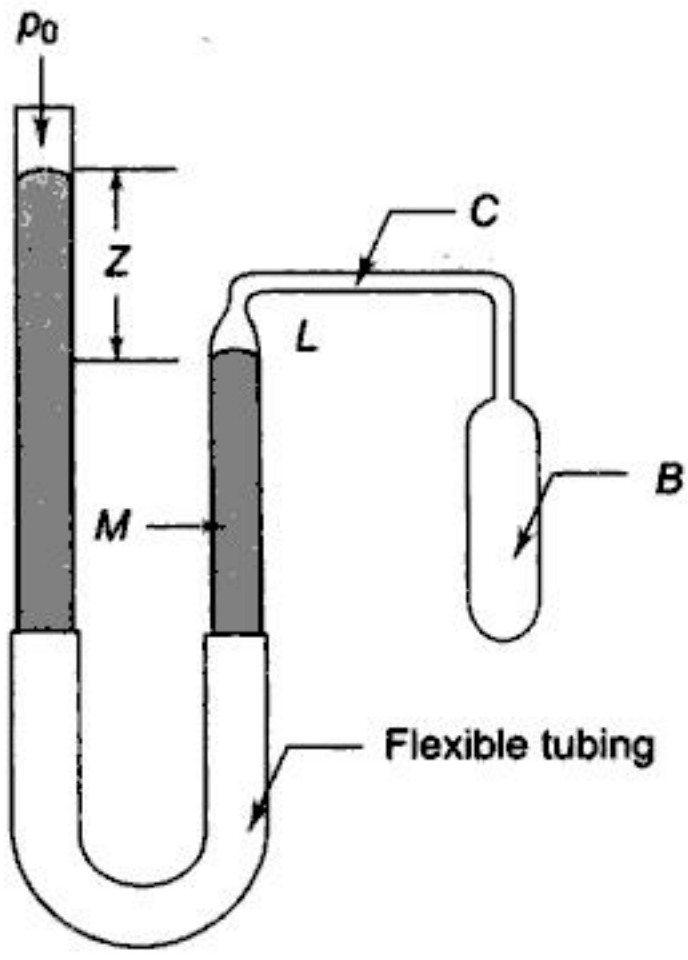

Fig. 2.1 *Constant Volume Gas Thermometer*

In a constant pressure gas thermometer, the mercury levels have to be adjusted to keep *Z* constant, and the volume of gas *V*, which would vary with the temperature of the system, becomes the thermometric property.

$$\therefore \qquad \Delta T = \frac{P}{R}\Delta V \qquad (2.11)$$

i.e. the temperature increase is proportional to the observed volume increase. The constant volume gas thermometer is, however, mostly in use, since it is simpler in construction and easier to operate.

2.6 IDEAL GAS TEMPERATURE

Let us suppose that the bulb of a constant volume gas thermometer contains an amount of gas such that when the bulb is surrounded by water at its triple point, the pressure p_t is 1000 mm Hg. Keeping the volume *V* constant, let the following procedure be conducted:

(a) Surround the bulb with steam condensing at 1 atm, determine the gas pressure *p* and calculate

$$\theta = 273.16 \frac{p}{1000}$$

(b) Remove some gas from the bulb so that when it is surrounded by water at its triple point, the pressure p_t is 500 mm Hg. Determine the new value of *p* and then θ for steam condensing at 1 atm.

$$\theta = 273.16 \frac{p}{500}$$

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(c) Continue reducing the amount of gas in the bulb so that p_t and p have smaller and smaller values, e.g. p_t having, say, 250 mm Hg, 100 mm Hg, and so on. At each value of p_t calculate the corresponding θ.

(d) Plot θ vs. p_t and extrapolate the curve to the axis where $p_t = 0$. Read from the graph

$$\lim_{p_t \to 0} \theta$$

The graph, as shown in Fig. 2.2, indicates that although the readings of a constant volume gas thermometer depend upon the nature of the gas, *all gases indicate the same temperature as p_t is lowered and made to approach zero.*

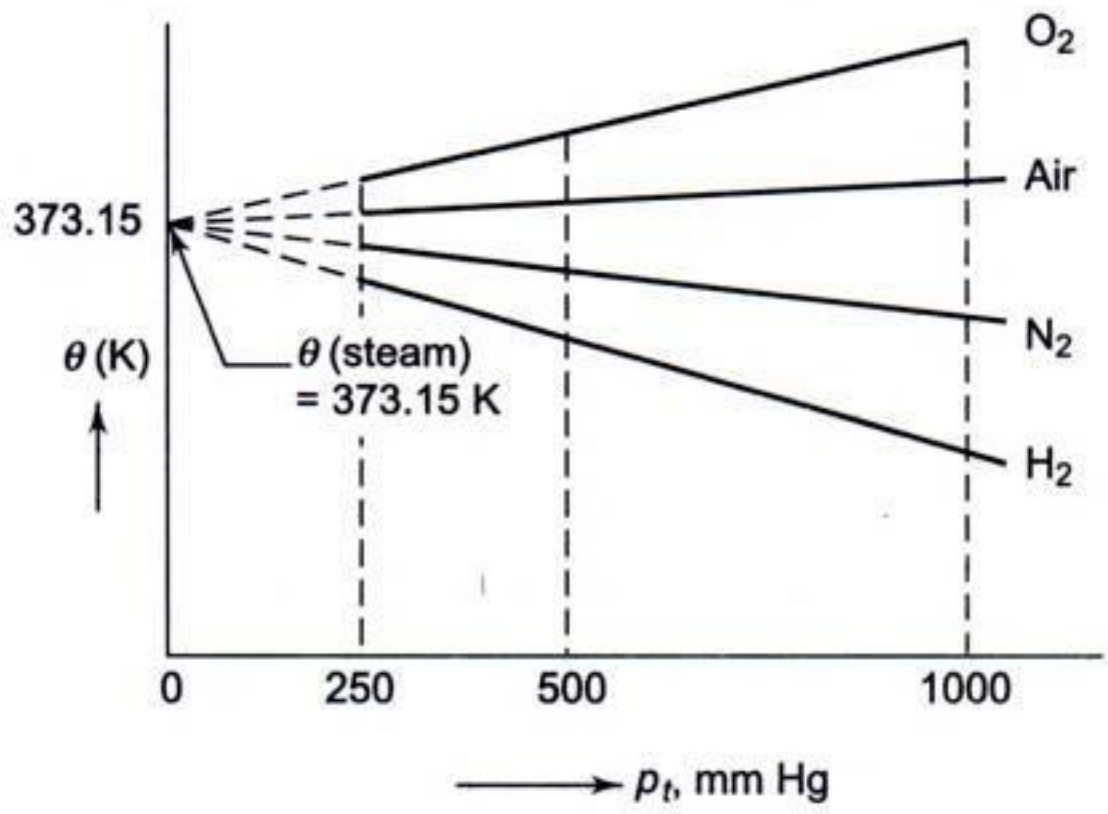

Fig. 2.2 *Ideal Gas Temperature for Steam Point*

A similar series of tests may be conducted with a constant pressure gas thermometer. The constant pressure may first be taken to be 1000 mm Hg, then 500 mm Hg, etc. and at each value of p, the volumes of gas V and V_t may be recorded when the bulb is surrounded by steam condensing at 1 atm and the triple point of water, respectively. The corresponding value of θ may be calculated from

$$\theta = 273.16 \frac{V}{V_t}$$

and θ vs. p may be plotted, similar to Fig. 2.2. It is found from the experiments that all gases indicate the same value of θ as p approaches zero.

Since a real gas, as used in the bulb, behaves as an ideal gas as pressure approaches zero (which would be explained later in Chapter 10), the *ideal gas temperature* T is defined by either of the two equations

$$T = 273.16 \lim_{p_t \to 0} \frac{p}{p_t}$$

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$$= 273.16 \lim_{p_t \to 0} \frac{V}{V_t} \tag{2.12}$$

where θ has been replaced by T to denote this particular temperature scale, the *ideal gas temperature scale*.

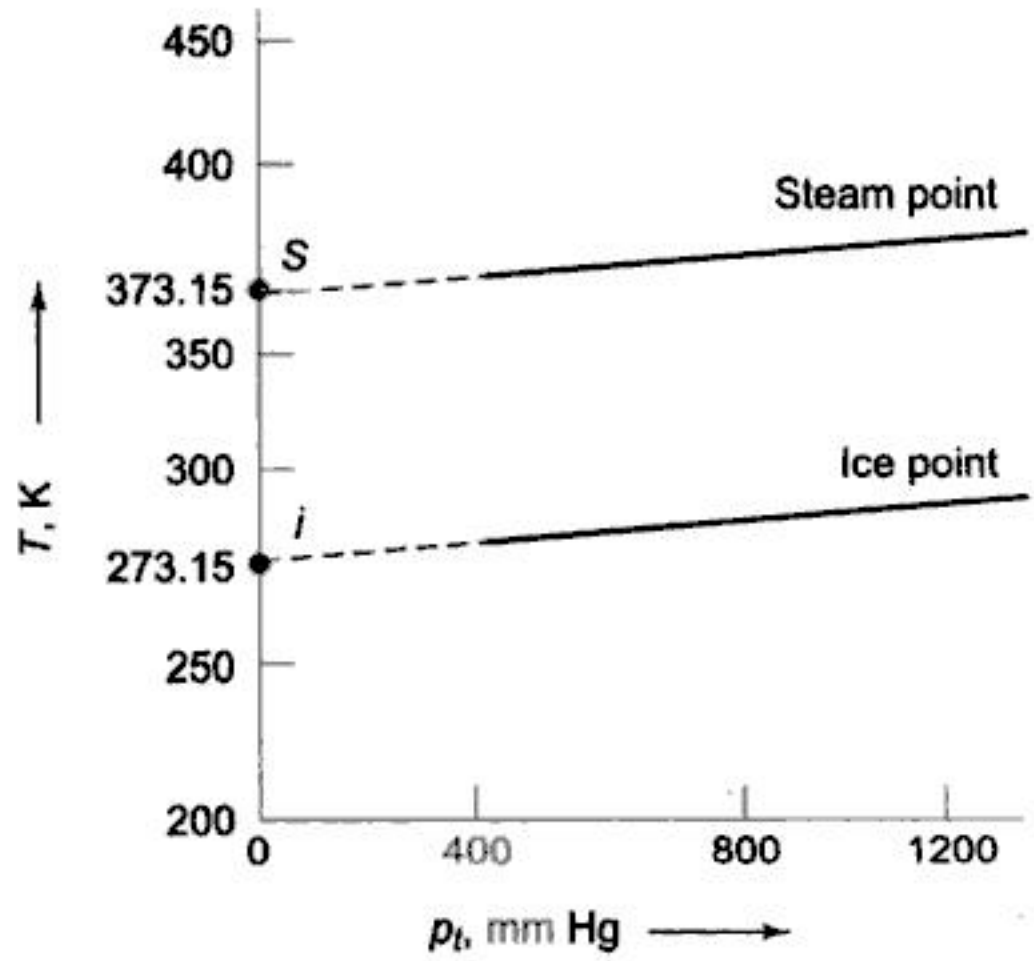

Fig. 2.3 *Steam-point and Ice-point from Constant Volume Gas Thermometer*

If p_s and p_t are the measured pressures at the steam point and the triple point respectively, one gets the value of the steam point temperature T_s as

$$T_s = 273.16 \lim_{p_t \to 0} \frac{p_s}{p_t} \tag{2.13}$$

which is equal to 373.15 K.

Similarly, the temperature T_i at the ice point is

$$T_i = 273.16 \lim_{p_t \to 0} \frac{p_i}{p_t} \tag{2.14}$$

which is equal to 273.15 K (Fig. 2.3).

Alternatively, if the ratio p_s/p_i is plotted against p_t with different gases in the bulb, one gets different curves as in Fig. 2.2. However, when extrapolated to zero pressure all curves converge, and the ratio p_s/p_i tends to a constant value giving

$$\frac{T_s}{T_i} = \lim_{p_t \to 0} \frac{p_s}{p_i} = 1.366099 \tag{2.15}$$

This value may be considered as a *universal constant*.

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One may now decide to have a certain number of divisions between the steam point and the ice point, say 100 as in the Kelvin and Celsius scales so that

$$T_s - T_i = 100 \tag{2.16}$$

Solving Eqs. (2.15) and (2.16),

$$T_s = 373.15 \text{ K and } T_i = 273.15 \text{ K}$$

2.7 CELSIUS TEMPERATURE SCALE

The Celsius temperature scale employs a degree of the same magnitude as that of the ideal gas scale, but its zero point is shifted, so that the Celsius temperature of the triple point of water is 0.01 degree Celsius or 0.01°C. If t denotes the Celsius temperature, then

$$t = T - 273.15°$$

Thus the Celsius temperature t_s at which steam condenses at 1 atm. pressure

$$t_s = T_s - 273.15°$$
$$= 373.15 - 273.15 = 100.00°C$$

Similar measurements for ice points show this temperature on the Celsius scale to be 0.00°C. The only Celsius temperature which is fixed by definition is that of the triple point.

2.8 ELECTRICAL RESISTANCE THERMOMETER

In the resistance thermometer (Fig. 2.4) the change in resistance of a metal wire due to its change in temperature is the thermometric property. The wire, frequently platinum, may be incorporated in a Wheatstone bridge circuit. The platinum resistance thermometer measures temperature to a high degree ofaccuracy and sensitivity, which makes it suitable as a standard for the calibration of other thermometers.

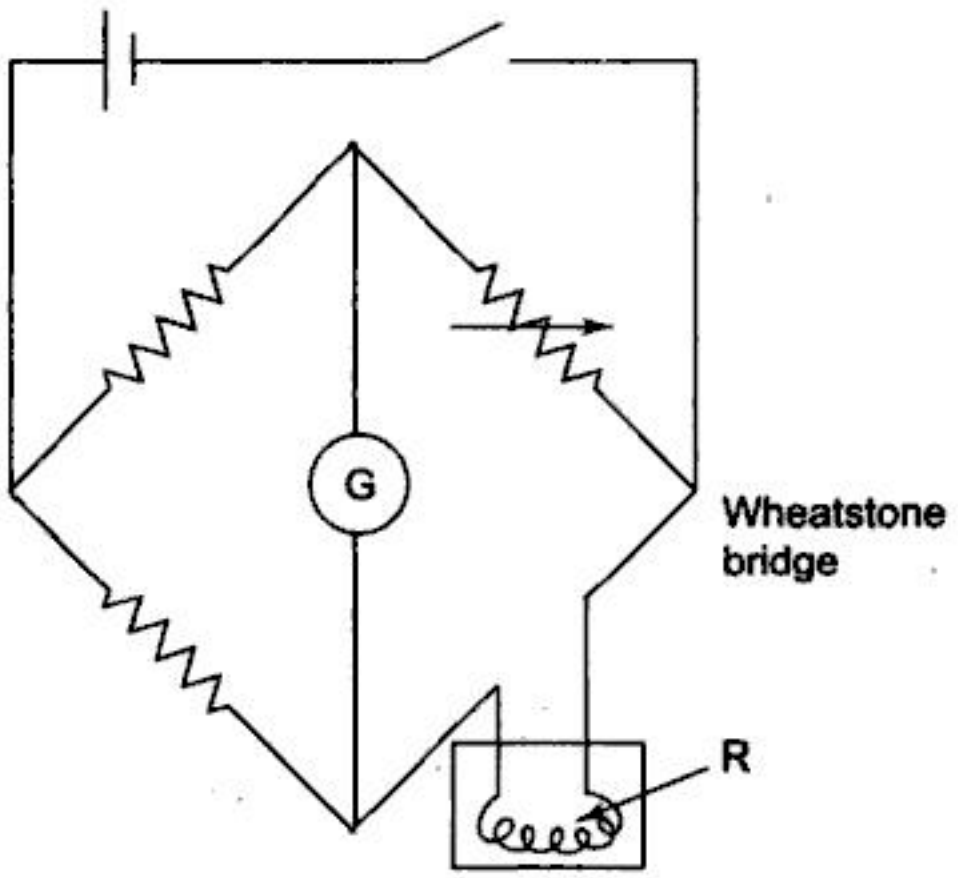

Fig. 2.4 *Resistance Thermometer*

In a restricted range, the following quadratic equation is often used

$$R = R_0 (1 + At + Bt^2)$$

where R_0 is the resistance of the platinum wire when it is surrounded by melting ice and A and B are constants.

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2.9 THERMOCOUPLE

A thermocouple circuit made up from joining two wires A and B made of dissimilar metals is shown in Fig. 2.5. Due to the Seebeck effect, a net e.m.f. is generated in the circuit which depends on the difference in temperature between the hot and cold junctions and is, therefore, a thermometric property of the circuit. This e.m.f. can be measured by a microvoltmeter to a high degree of accuracy. The choice of metals depends largely on the temperature range to be investigated, and copper-constantan, chromel-alumel and platinum-platinum-rhodium are typical combinations in use.

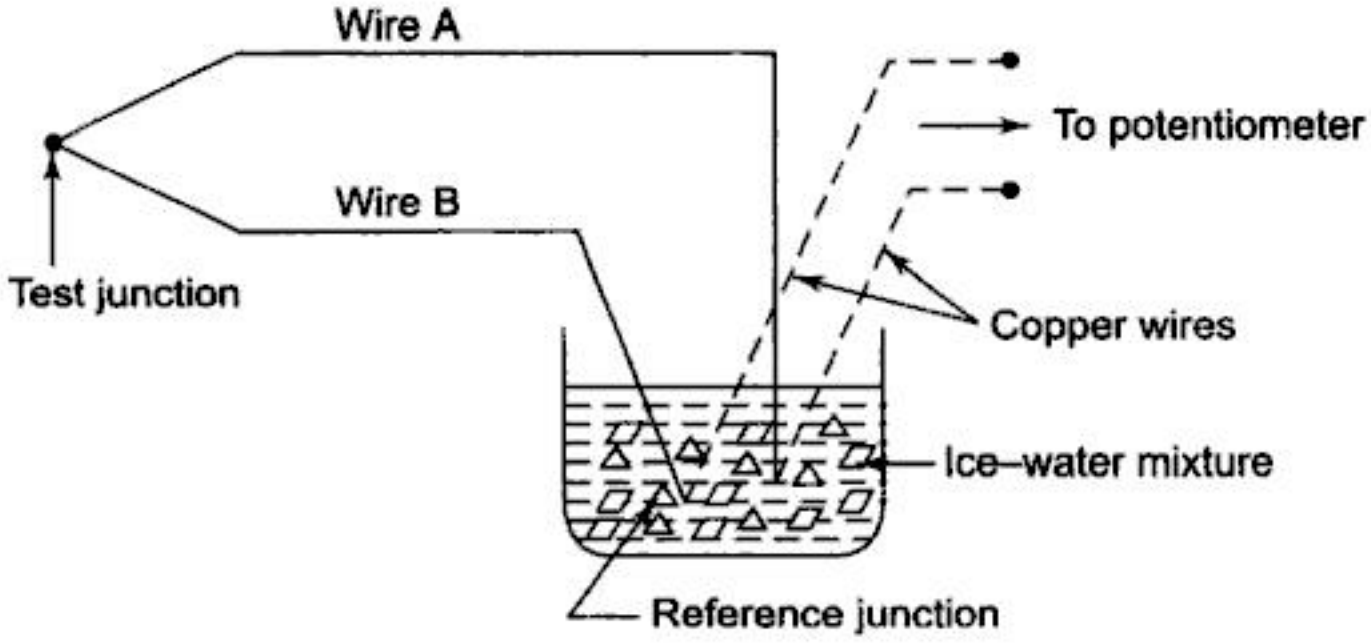

Fig. 2.5 *Thermocouple*

A thermocouple is calibrated by measuring the thermal e.m.f. at various known temperatures, the reference junction being kept at 0°C. The results of such measurements on most thermocouples can usually be represented by a cubic equation of the form

$$\varepsilon = a + bt + ct^2 + dt^3$$

where ε is the thermal e.m.f. and the constants a, b, c and d are different for each thermocouple.

The advantage of a thermocouple is that it comes to thermal equilibrium with the system, whose temperature is to be measured, quite rapidly, because its mass is small.

2.10 INTERNATIONAL PRACTICAL TEMPERATURE SCALE

An international temperature scale was adopted at the Seventh General Conference on Weights and Measures held in 1927. It was not to replace the Celsius or ideal gas scales, but to provide a scale that could be easily and rapidly used to calibrate scientific and industrial instruments. Slight refinements were incorporated into the scale in revisions adopted in 1948, 1954, 1960 and 1968. The international practical scale agrees with the Celsius scale at the defining fixed points listed in Table 2.2. The temperature interval from the oxygen point to the gold point is divided into three main parts, as given below.

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Table 2.2 *Temperatures of Fixed Points*

	Temperature °C
Normal boiling point of oxygen	– 182.97
Triple point of water (standard)	+ 0.01
Normal boiling point of water	100.00
Normal boiling point of sulphur	444.60
(Normal melting point of zinc-suggested as an alternative to the sulphur point)	419.50
Normal melting point of antimony	630.50
Normal melting point of silver	960.80
Normal melting point of gold	1063.00

(a) From 0 to 660°C A platinum resistance thermometer with a platinum wire whose diameter must lie between 0.05 and 0.20 mm is used, and the temperature is given by the equation

$$R = R_0 (1 + At + Bt^2)$$

where the constants R_0, A, and B are computed by measurements at the ice point, steam point, and sulphur point.

(b) From –190 to 0°C The same platinum resistance thermometer is used, and the temperature is given by

$$R = R_0 [1 + At + Bt^2 + C (t - 100) t^3]$$

where R_0, A and B are the same as before, and C is determined from a measurement at the oxygen point.

(c) From 660 to 1063°C A thermocouple, one wire of which is made of platinum and the other of an alloy of 90% platinum and 10% rhodium, is used with one junction at 0°C. The temperature is given by the formula

$$\varepsilon = a + bt + ct^2$$

where a, b, and c are computed from measurements at the antimony point, silver point, and gold point. The diameter of each wire of the thermocouple must lie between 0.35 and 0.65 mm.

An optical method is adopted for measuring temperatures higher than the gold point. The intensity of radiation of any convenient wavelength is compared with the intensity of radiation of the same wavelength emitted by a black body at the gold point. The temperature is then determined with the help of Planck's law of thermal radiation.

Solved Examples

Example 2.1 Two mercury-in-glass thermometers are made of identical materials and are accurately calibrated at 0°C and 100°C. One has a tube of

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constant diameter, while the other has a tube of conical bore, ten per cent greater in diameter at 100°C than at 0°C. Both thermometers have the length between 0 and 100 subdivided uniformly. What will be the straight bore thermometer read in a place where the conical bore thermometer reads 50°C?

Solution The volume of mercury in the tube at t°C, V_t, is given by

$$V_t = V_0 [1 + \beta (t - t_0)]$$

where V_0 is the volume of mercury at 0°C, β is the coefficient of volume expansion of mercury, and t_0 is the ice point temperature which is 0°C. The volume change of glass is neglected.

Therefore $$V_t - V_0 = \beta V_0 t$$

The temperature t is thus a linear function of volume change of mercury $(V_t - V_0)$.

Therefore $$\Delta V_{0-100} = \beta V_0 \cdot 100$$

$$\Delta V_{0-50} = \beta V_0 \cdot 50$$

$$\therefore \quad \frac{\Delta V_{0-50}}{\Delta V_{0-100}} = \frac{1}{2}$$

i.e. at 50°C, the volume of mercury will be half of that at 100°C, for the straight bore thermometer [Fig. 2.6(a)].

But if the bore is conical [Fig. 2.6 (b)], mercury will fill up the volume ACDB, which is less than half of the mercury volume at 100°C, i.e. volume AEFB. Let t be the true temperature when mercury rises half the length of the conical tube (the apparent temperature being 50°C). Let EA and FB be extended to meet at G. Let l represent the length of the thermometers and l' the vertical height of the cone ABG, as shown in the figure. Now,

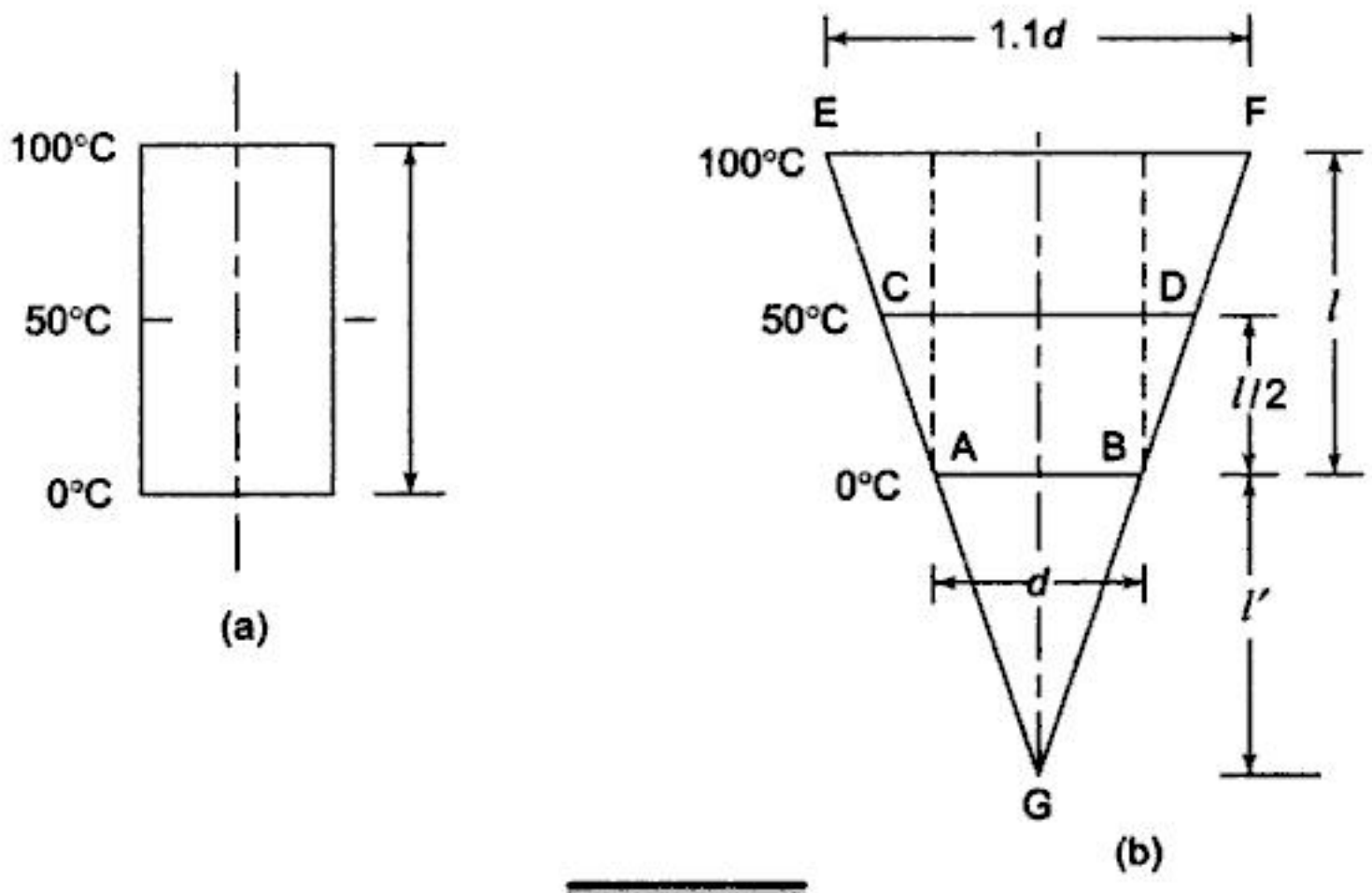

Fig. 2.6

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$$\frac{l'}{l+l'} = \frac{d}{1.1d} = \frac{1}{1.1}$$

$$l' = 10$$

and $$\frac{l'}{l'+l/2} = \frac{d}{CD}$$

$\therefore$ $$CD = \frac{10.5}{10}\, d = 1.05d$$

Again

$$\Delta V_{0-100} = V_0 \cdot \beta \cdot 100$$

$$\Delta V_{0-t} = V_0 \beta t$$

$$\frac{\Delta V_{0-t}}{\Delta V_{0-100}} = \frac{t}{100}$$

or $$\frac{\text{Volume ACDB}}{\text{Volume AEFB}} = \frac{t}{100}$$

or $$\frac{\frac{1}{3}\frac{\pi}{4}(1.05d)^2 \times 10.5l - \frac{1}{3}\frac{\pi}{4}d^2 \cdot 10l}{\frac{1}{3}\frac{\pi}{4}(1.1d)^2 \times 11l - \frac{1}{3}\frac{\pi}{4}d^2 \cdot 10l} = \frac{t}{100}$$

or $$\frac{1.05 \times 1.05 \times 10.5 - 10}{1.1 \times 1.1 \times 11 - 10} = \frac{t}{100}$$

$\therefore$ $$t = \frac{1.58}{3.31} \times 100 = \mathbf{47.7°C}$$

Example 2.2 The e.m.f. in a thermocouple with the test junction at t°C on gas thermometer scale and reference junction at ice point is given by

$$\varepsilon = 0.20\, t - 5 \times 10^{-4}\, t^2 \text{ mV}$$

The millivoltmeter is calibrated at ice and steam points. What will this thermometer read in a place where the gas thermometer reads 50°C?

Solution At ice point, when $t = 0°C$, $\varepsilon = 0$ mV

At steam point, when $t = 100°C$, $\varepsilon = 0.20 \times 100 - 5 \times 10^{-4} \times (100)^2$

$= 15$ mV

At $t = 50°C$, $\varepsilon = 0.20 \times 50 - 5 \times 10^{-4}\,(50)^2 = 8.75$ mV

When the gas thermometer reads 50°C, the thermocouple will read

$$\frac{100}{15} \times 8.75, \text{ or } \mathbf{58.33°C}$$

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Example 2.3 A platinum resistance thermometer has a resistance of 2.8 ohm at 0°C and 3.8 ohm at 100°C. Calculate the temperature when the resistance indicated is 5.8 ohm.

Solution Let $R = R_0 (1 + \alpha t)$
where R_0 is the resistance at 0°C.

Therefore, $R_0 = 2.8$ ohm

$$R_{100} = 3.8 \text{ ohm} = 2.8 (1 + \alpha \times 100)$$

$$\therefore \quad \alpha = \left(\frac{3.8}{2.8} - 1\right) \times 10^{-2} = 0.357 \times 10^{-2}$$

When $R = 5.8$ ohm,

$$5.8 = 2.8 (1 + 0.357 \times 10^{-2}\, t)$$

or, $$t = \left(\frac{5.8}{2.8} - 1\right) \times \frac{100}{0.357} = \mathbf{300°C.}$$

Review Questions

2.1 What is the zeroth law of thermodynamics?
2.2 Define thermometric property.
2.3 What is a thermometer?
2.4 What is a fixed point?
2.5 How many fixed points were used prior to 1954? What are these?
2.6 What is the standard fixed point in thermometry? Define it.
2.7 Why is a gas chosen as the standard thermometric substance?
2.8 What is an ideal gas?
2.9 What is the difference between the universal gas constant and a characteristic gas constant?
2.10 What is a constant volume gas thermometer? Why is it preferred to a constant pressure gas thermometer?
2.11 What do you understand by the ideal gas temperature scale?
2.12 How can the ideal gas temperature for the steam point be measured?
2.13 What is the Celsius temperature scale?
2.14 What is the advantage of a thermocouple in temperature measurement?
2.15 How does the resistance thermometer measure temperature?
2.16 What is the need of the international practical temperature scale?

Problems

2.1 The limiting value of the ratio of the pressure of gas at the steam point and at the triple point of water when the gas is kept at constant volume is found to be 1.36605. What is the ideal gas temperature of the steam point? **Ans. 100°C**

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2.2 In a constant volume gas thermometer the following pairs of pressure readings were taken at the boiling point of water and the boiling point of sulphur, respectively:

Water b.p.	50.0	100	200	300
Sulphur b.p.	96.4	193	387	582

The numbers are the gas pressures, mm Hg, each pair being taken with the same amount of gas in the thermometer, but the successive pairs being taken with different amounts of gas in the thermometer. Plot the ratio of $S_{b.p.}$:$H_2O_{b.p.}$ against the reading at the water boiling point, and extrapolate the plot to zero pressure at the water boiling point. This gives the ratio of $S_{b.p.}$:$H_2O_{b.p.}$ on a gas thermometer operating at zero gas pressure, i.e., an ideal gas thermometer. What is the boiling point of sulphur on the gas scale, from your plot? Ans. 445°C

2.3 The resistance of a platinum wire is found to be 11,000 ohms at the ice point, 15.247 ohms at the steam point, and 28.887 ohms at the sulphur point. Find the constants A and B in the equation

$$R = R_0 (1 + At + Bt^2)$$

and plot R against t in the range 0 to 660°C.

2.4 When the reference junction of a thermocouple is kept at the ice point and the test junction is at the Celsius temperature t, and e.m.f. e of the thermocouple is given by the equation

$$\varepsilon = at + bt^2$$

where $a = 0.20$ mV/deg, and $b = -5.0 \times 10^{-4}$ mV/deg^2

(a) Compute the e.m.f. when $t = -100$°C, 200°C, 400°C, and 500°C, and draw graph of ε against t in this range.

(b) Suppose the e.m.f. ε is taken as a thermometric property and that a temperature scale t^* is defined by the linear equation.

$$t^* = a' \varepsilon + b'$$

and that $t^* = 0$ at the ice point and $t^* = 100$ at the steam point. Find the numerical values of a' and b' and draw a graph of ε against t^*.

(c) Find the values of t^* when $t = -100$°C, 200°C, 400°C, and 500°C, and draw a graph of t^* against t.

(d) Compare the Celsius scale with the t^* scale.

2.5 The temperature t on a thermometric scale is defined in terms of a property K by the relation

$$t = a \ln K + b$$

where a and b are constants.

The values of K are found to be 1.83 and 6.78 at the ice point and the steam point, the temperatures of which are assigned the numbers 0 and 100 respectively. Determine the temperature corresponding to a reading of K equal to 2.42 on the thermometer. Ans. 21.346°C

2.6 The resistance of the windings in a certain motor is found to be 80 ohms at room temperature (25°C). When operating at full load under steady state conditions, the motor is switched off and the resistance of the windings, immediately measured again, is found to be 93 ohms. The windings are made of copper whose resistance at temperature t°C is given by

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$$R_t = R_0 [1 + 0.00393\ t]$$

where R_0 is the resistance at 0°C. Find the temperature attained by the coil during full load.

Ans. 70.41°C

2.7 A new scale N of temperature is divided in such a way that the freezing point of ice is 100°N and the boiling point is 400°N. What is the temperature reading on this new scale when the temperature is 150°C? At what temperature both the Celsius and the new temperature scale reading would be the same?

Ans. 550°N, – 50°C.

2.8 A platinum wire is used as a resistance thermometer. The wire resistance was found to be 10 ohm and 16 ohm at ice point and steam point respectively, and 30 ohm at sulphur boiling point of 444.6°C. Find the resistance of the wire at 500°C, if the resistance varies with temperature by the relation.

$$R = R_0 (1 + \alpha t + \beta t^2)$$

Ans. 31.3 ohm

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CHAPTER 3

Work and Heat Transfer

A closed system and its surroundings can interact in two ways: (a) by work transfer, and (b) by heat transfer. These may be called *energy interactions* and these bring about changes in the properties of the system. Thermodynamics mainly studies these energy interactions and the associated property changes of the system.

3.1 WORK TRANSFER

Work is one of the basic modes of energy transfer. In mechanics the action of a force on a moving body is identified as work. A force is a means of transmitting an effect from one body to another. But a force itself never produces a physical effect except when coupled with motion and hence it is not a form of energy. An effect such as the raising of a weight through a certain distance can be performed by using a small force through a large distance or a large force through a small distance. The product of force and distance is the same to accomplish the same effect. In mechanics work is defined as:

The work is done by a force as it acts upon a body moving in the direction of the force.

The action of a force through a distance (or of a torque through an angle) is called *mechanical work* since other forms of work can be identified, as discussed later. The product of the force and the distance moved parallel to the force is the magnitude of mechanical work.

In thermodynamics, work transfer is considered as occurring between the system and the surroundings. *Work is said to be done by a system if the sole effect on things external to the system can be reduced to the raising of a weight.* The weight may not actually be raised, but the net effect external to the system would be the raising of a weight. Let us consider the battery and the motor in Fig. 3.1 as a system. The motor is driving a fan. The system is doing work upon the surroundings. When the fan is replaced by a pulley and a weight, as shown in Fig. 3.2, the weight may be raised with the pulley driven by the motor. The sole effect on things external to the system is then the raising of a weight.

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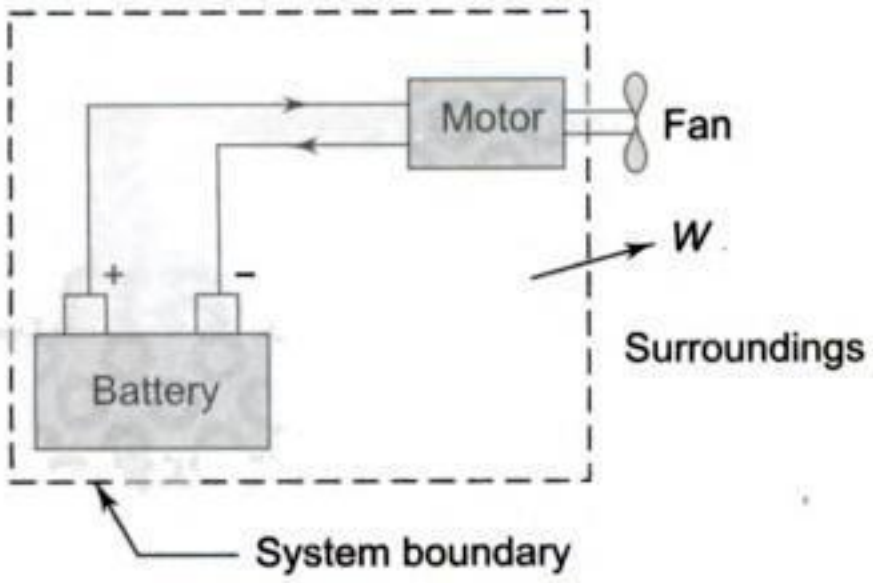

Fig. 3.1 *Battery-motor System Driving a Fan*

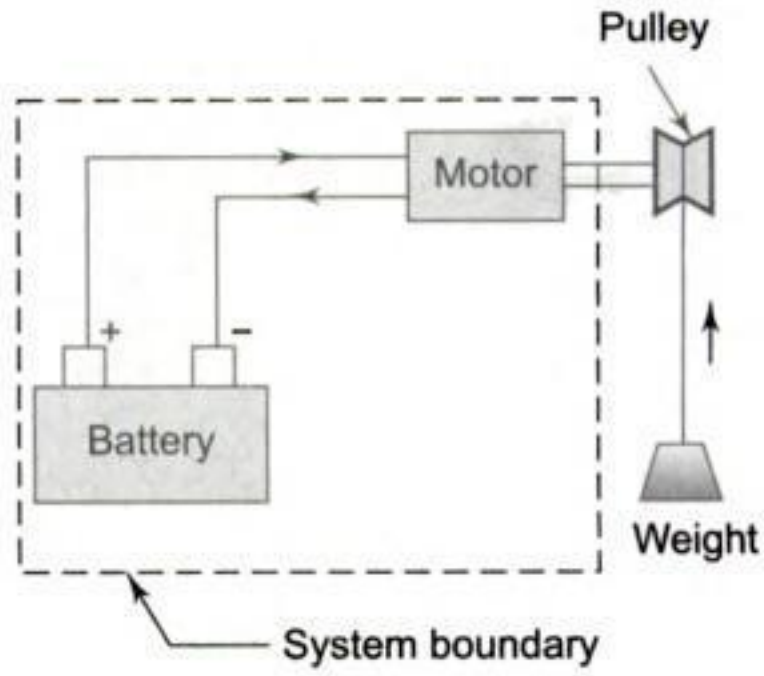

Fig. 3.2 *Work Transfer from a System*

When work is done by a system, it is arbitrarily taken to be positive, and when work is done on a system, it is taken to be negative (Fig. 3.3). The symbol W is used for work transfer.

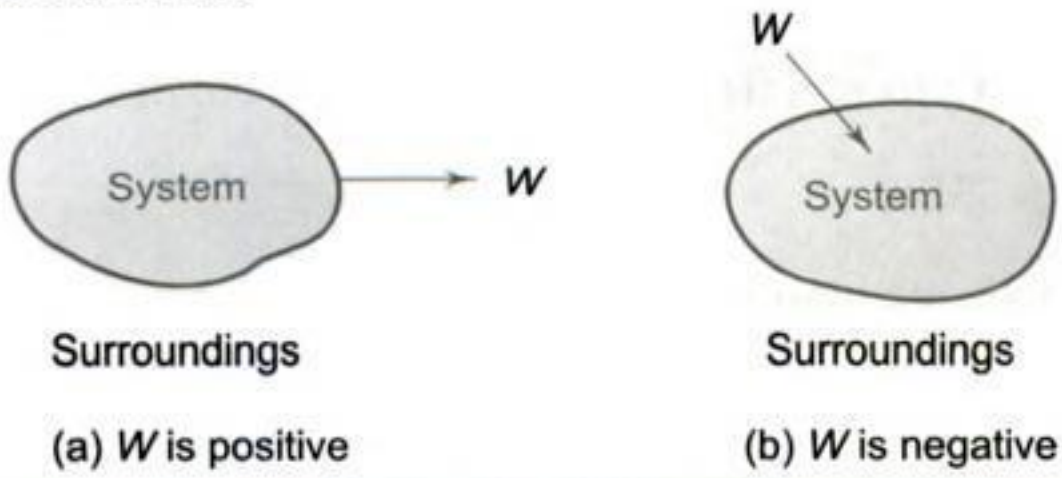

Fig. 3.3 *Work Interaction between a System and the Surroundings*

The unit of work is N.m or Joule [1 Nm = 1 Joule]. The rate at which work is done by, or upon, the system is known as *power*. The unit of power is J/s or watt.

Work is one of the forms in which a system and its surroundings can interact with each other. There are various types of work transfers which can get involved between them.

3.2 *pdV*-WORK OR DISPLACEMENT WORK

Let the gas in the cylinder (Fig. 3.4) be a system having initially the pressure p_1 and volume V_1. The system is in thermodynamic equilibrium, the state of which is described by the coordinates p_1, V_1. The piston is the only boundary which moves due to gas pressure. Let the piston move out to a new final position 2, which is also a thermodynamic equilibrium state specified by pressure p_2 and volume V_2. At any intermediate point in the travel of the piston, let the pressure be p and the volume V. This must also be an equilibrium state,

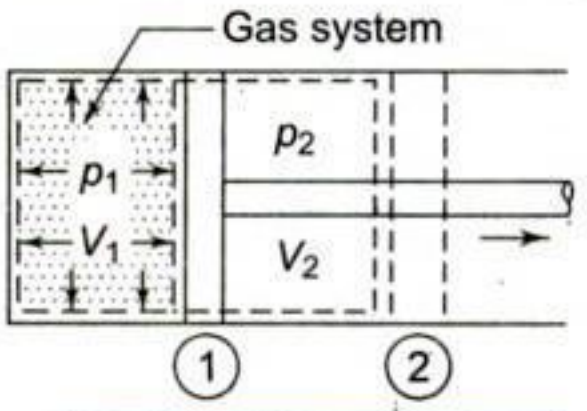

Fig. 3.4 *pdV Work*

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since macroscopic properties p and V are significant only for equilibrium states. When the piston moves an infinitesimal distance dl, and if 'a' be the area of the piston, the force F acting on the piston $F = p.a.$ and the infinitesimal amount of work done by the gas on the piston

$$dW = F \cdot dl = padl = pdV \tag{3.1}$$

where $dV = adl$ = infinitesimal displacement volume. The differential sign in $đ W$ with the line drawn at the top of it will be explained later.

When the piston moves out from position 1 to position 2 with the volume changing from V_1 to V_2, the amount of work W done by the system will be

$$W_{1-2} = \int_{V_1}^{V_2} pdV$$

The magnitude of the work done is given by the area under the path 1–2, as shown in Fig. 3.5. Since p is at all times a thermodynamic coordinate, all the states passed through by the system as the volume changes from V_1 to V_2 must be equilibrium states, and the path 1--2 *must be quasi-static*. The piston moves infinitely slowly so that every state passed through is an equilibrium state. *The integration* $\int pdV$ *can be performed only on a quasi-static path.*

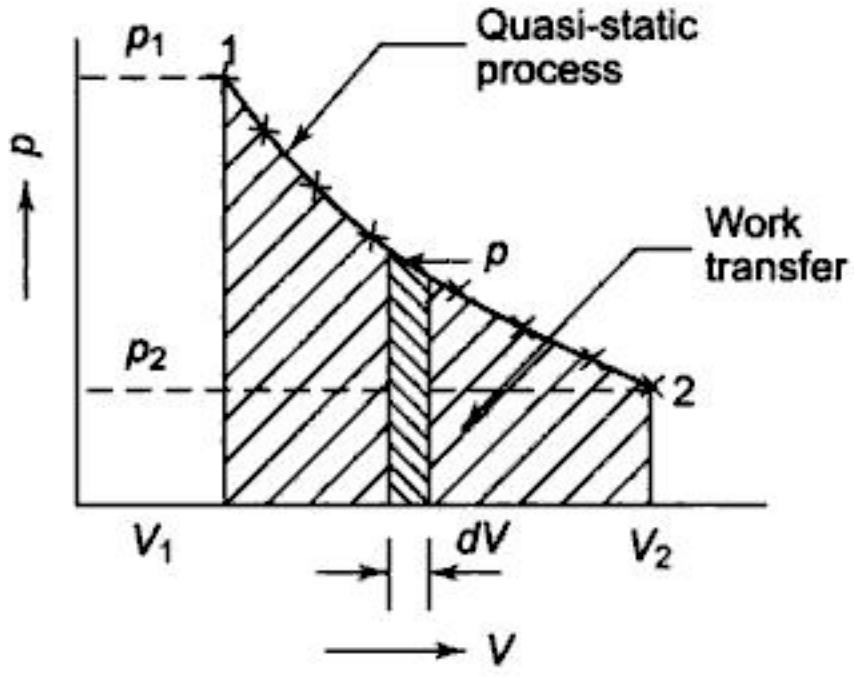

Fig. 3.5 *Quasi-Static pdV Work*

3.2.1 Path Function and Point Function

With reference to Fig. 3.6, it is possible to take a system from state 1 to state 2 along many quasi-static paths, such as A, B or C. Since the area under each curve represents the work for each process, the amount of work involved in each case is not a function of the end states of the process, and it depends on the path the system follows in going from state 1 to state 2. For this reason, work is called a *path function*, and $đ W$ is an *inexact or imperfect differential.*

Thermodynamic properties are *point functions*, since for a given state, there is a definite value for each property. The change in a thermodynamic property of a system in a change of state is independent of the path the system follows during the change of state, and depends only on the initial and final states of the system. The differentials of point functions are *exact or perfect differentials*, and the integration is simply

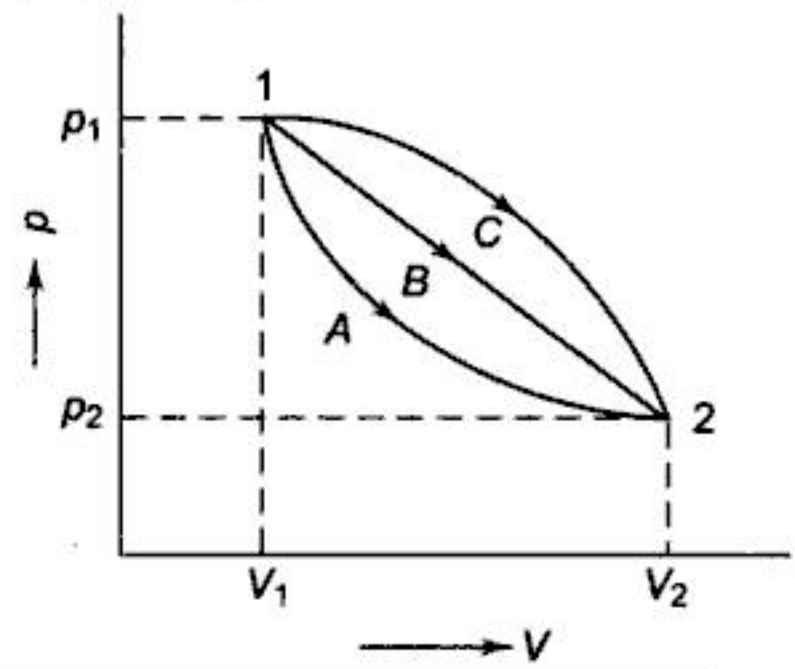

Fig. 3.6 *Work—A Path Function*

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$$\int_{V_1}^{V_2} dV = V_2 - V_1$$

The change in volume thus depends only on the end states of the system irrespective of the path the system follows.

On the other hand, work done in a quasi-static process between two given states depends on the path followed.

$$\int_1^2 đ W \neq W_2 - W_1$$

Rather,

$$\int_1^2 đ W = W_{1-2} \qquad \text{or} \qquad {}_1W_2$$

To distinguish an inexact differential $đ W$ from an exact differential dV or dp the differential sign is being cut by a line at its top.

From Eq. (3.1),

$$dV = \frac{1}{p} đ W \tag{3.2}$$

Here, $1/p$ is called the *integrating factor*. Therefore, an inexact differential $đ W$ when multiplied by an integrating factor $1/p$ becomes an exact differential dV.

For a cyclic process, the initial and final states of the system are the same, and hence, the change in any property is zero, i.e.

$$\oint dV = 0, \quad \oint dp = 0, \quad \oint dT = 0 \tag{3.3}$$

where the symbol $\oint$ denotes the cyclic integral for the closed path. Therefore, *the cyclic integral of a property is always zero.*

3.2.2 *pdV*-Work in Various Quasi-Static Processes

(a) Constant pressure process (Fig. 3.7) (isobaric or isopiestic process)

$$W_{1-2} = \int_{V_1}^{V_2} pdV = p(V_2 - V_1) \tag{3.4}$$

(b) Constant volume process (Fig. 3.8) (isochoric process)

$$W_{1-2} = \int pdV = 0 \tag{3.5}$$

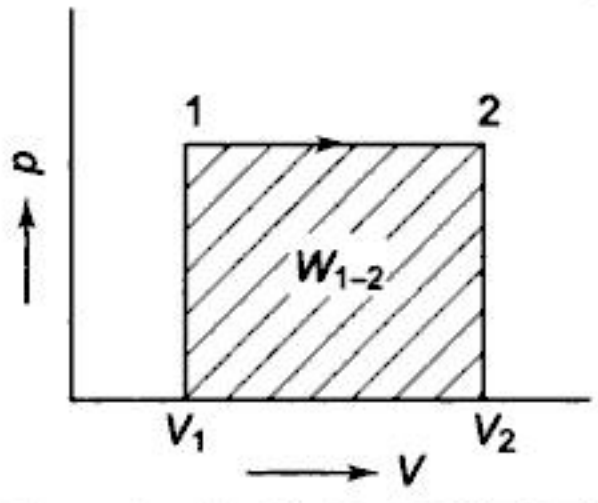

Fig. 3.7 *Constant Pressure Process*

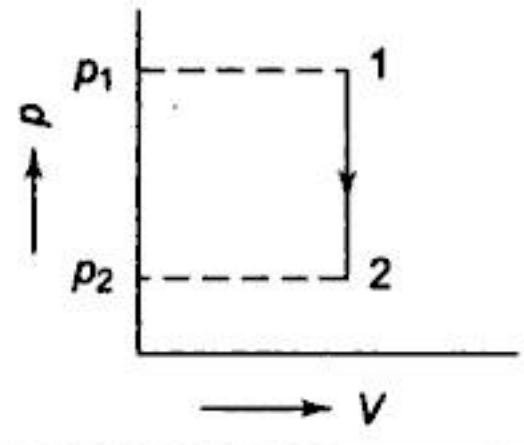

Fig. 3.8 *Constant Volume Process*

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(c) Process in which $pV = C$ (Fig. 3.9)

$$\therefore \qquad W_{1-2} = \int_{V_1}^{V_2} p\,dV \qquad pV = p_1 V_1 = C$$

$$p = \frac{(p_1 V_1)}{V}$$

$$W_{1-2} = p_1 V_1 \int_{V_1}^{V_2} \frac{dV}{V} = p_1 V_1 \ln \frac{V_2}{V_1}$$

$$= p_1 V_1 \ln \frac{p_1}{p_2} \qquad (3.6)$$

(d) Process in which $pV^n = C$, where n is a constant (Fig. 3.10).

$$pV^n = p_1 V_1^n = p_2 V_2^n = C$$

$$\therefore \qquad p = \frac{\left(p_1 V_1^n\right)}{V^n}$$

$$\therefore \qquad W_{1-2} = \int_{V_1}^{V_2} p\,dV = \int_{V_1}^{V_2} \frac{p_1 V_1^n}{V^n} \cdot dV = (p_1 V_1^n) \left[\frac{V^{-n+1}}{-n+1}\right]_{V_1}^{V_2}$$

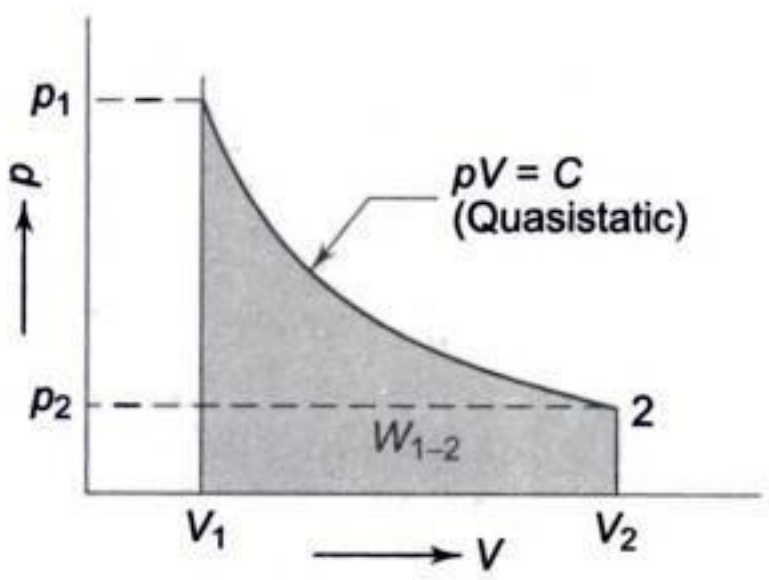

Fig. 3.9 *Process in Which pV = Constant*

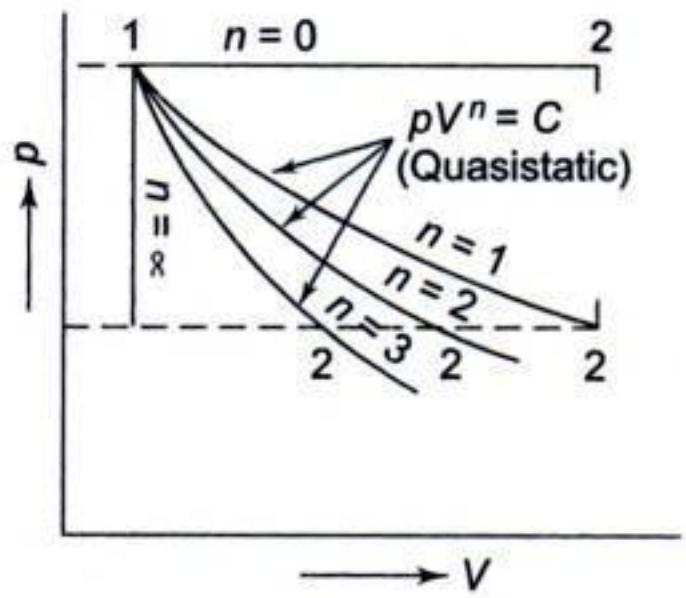

Fig. 3.10 *Process in which pV^n = Constant*

$$= \frac{p_1 V_1^n}{1-n} \left(V_2^{1-n} - V_1^{1-n}\right) = \frac{p_2 V_2^n \times V_2^{1-n} - p_1 V_1^n \times V_1^{1-n}}{1-n}$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1} = \frac{p_1 V_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{n-1/n}\right] \qquad (3.7)$$

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3.3 INDICATOR DIAGRAM

An indicator diagram is a trace made by a recording pressure gauge, called the indicator, attached to the cylinder of a reciprocating engine. This represents the work done in one engine cycle. Figure 3.11 shows a typical engine indicator.

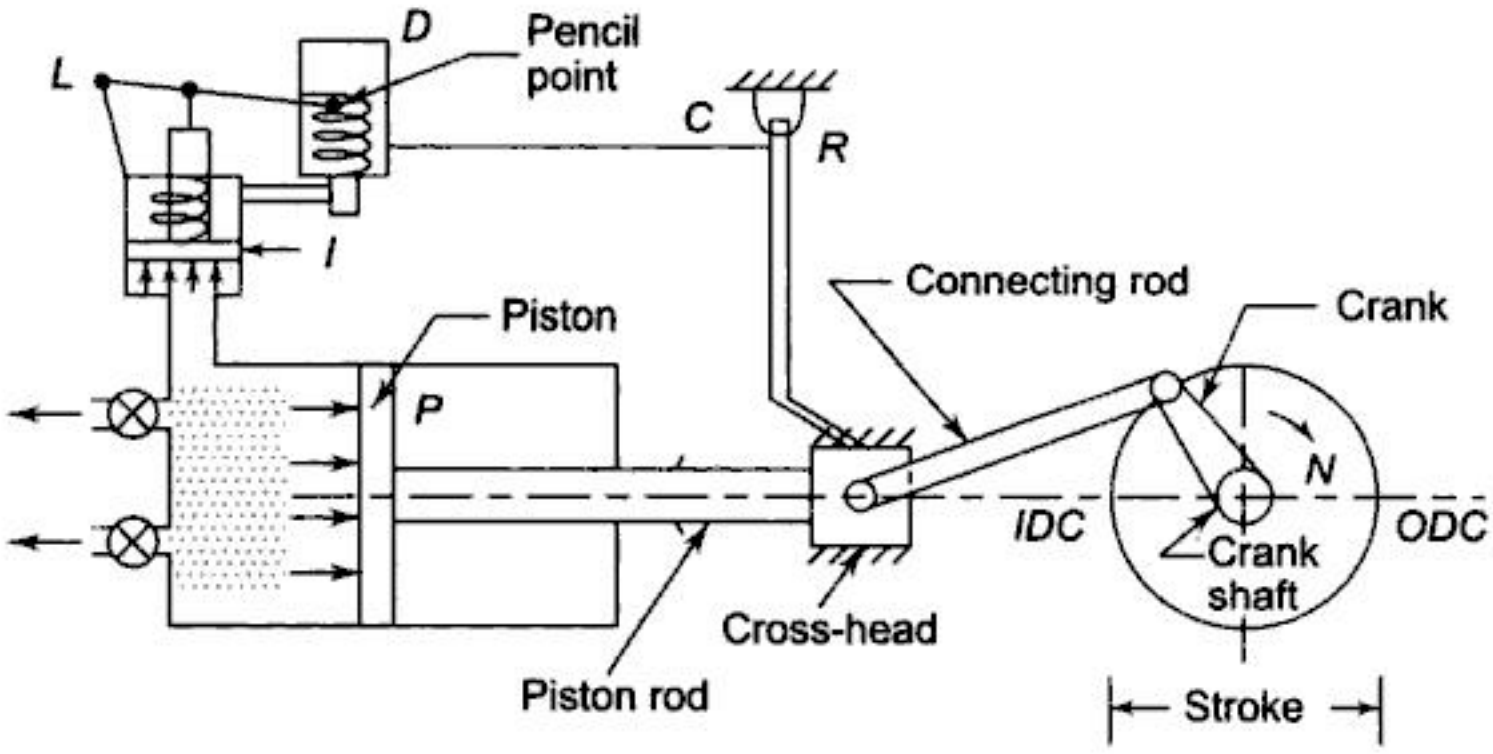

Fig. 3.11 *Engine Indicator*

The same gas pressure acts on both the engine piston P and the indicator piston I. The indicator piston is loaded by a spring and it moves in direct proportion to the change in pressure. The motion of the indicator piston causes a pencil held at the end of the linkage L to move upon a strip of paper wrapped around drum D. The drum is rotated about its axis by cord C, which is connected through a reducing motion R to the piston P of the engine. The surface of drum D moves horizontally under the pencil while the pencil moves vertically over the surface and a plot of pressure upon the piston vs. piston travel is obtained.

Before tracing the final indicator diagram, a pressure reference line is recorded by subjecting the indicator to the atmosphere and tracing a line at a constant pressure of one atmosphere.

The area of the indicator diagram represents the magnitude of the net work done by the system in one engine cycle. The area under the path 1–2 represents work done by the system and the area under the path 2–1 represents work done upon the system (Fig. 3.12). The area of the diagram, a_d, is measured by means of a planimeter, and the length of the diagram, l_d, is also measured. The *mean effective pressure* (m.e.p.) p_m is defined in the following way

$$p_m = \frac{a_d}{l_d} \times K$$

where K is the indicator spring constant (N/cm^2 × cm travel). Work done in one engine cycle

$$= (p_m \cdot A)\, L$$

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where A = cross-sectional area of the cylinder

$= \frac{\pi}{4} D^2$, where D is the cylinder diameter

and L = stroke of piston, or length of cylinder.

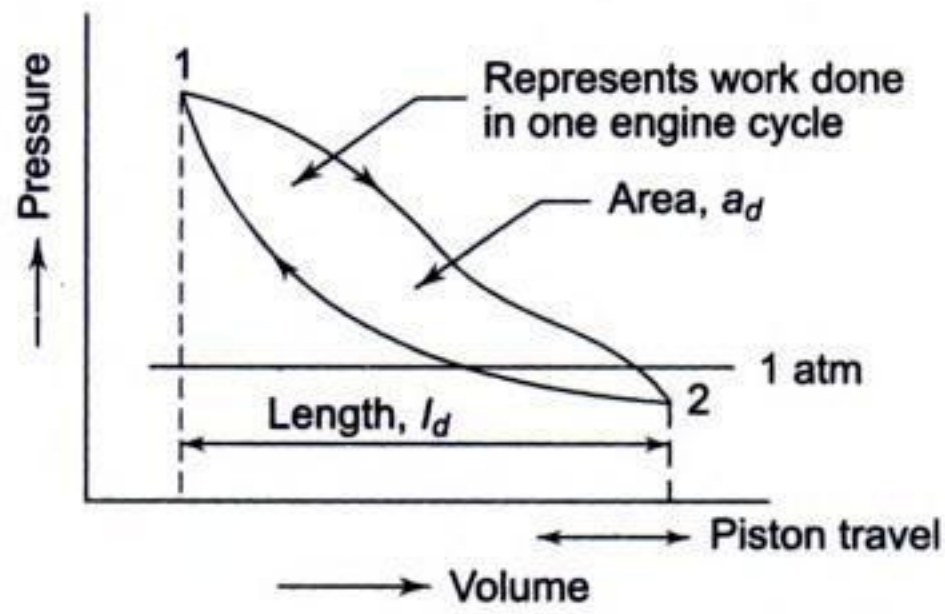

Fig. 3.12 *Indicator Diagram*

Let N be the revolutions per minute (r.p.m.) of the crankshaft. In a two-stroke cycle, the engine cycle is completed in two strokes of the piston or in one revolution of the crankshaft. In a four-stroke cycle, the engine cycle is compieted in four strokes of the piston or two revolutions of the crankshaft.

For a two-stroke engine, work done in one minute = $p_m ALN$, and for a four-stroke engine, work done in one minute = $p_m ALN/2$.

The power developed inside the cylinder of the engine is called *indicated power* (IP),

$$\therefore \quad \text{IP} = \frac{p_m AL\left(N \text{ or } \frac{N}{2}\right) n}{60} \text{ kW} \tag{3.8}$$

where p_m is in kPa and n is the number of cylinders in the engine.

The power available at the crankshaft is always less than this value (IP) due to friction, etc. and is called the *brake power* (BP) or *shaft power* (SP). If w is the angular velocity of the crankshaft in radian/sec, then

$$\text{BP} = T\,\omega \tag{3.9}$$

where T is the torque transmitted to the crankshaft in mN.

$$\therefore \quad \text{BP} = \frac{2\pi\, TN}{60} \tag{3.10}$$

where N is the number of revolutions per minute (rpm).

The mechanical efficiency of the engine, η_{mech}, is defined as

$$\eta_{\text{mech}} = \frac{\text{BP}}{\text{IP}} \tag{3.11}$$

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An engine is said to be *double-acting*, if the working fluid is made to work on both sides of the piston. Such an engine theoretically develops twice the amount of work developed in a single-acting engine. Most reciprocating steam engines are double-acting, and so are many marine diesel engines. Internal combustion engines for road transport are always single-acting.

3.4 OTHER TYPES OF WORK TRANSFER

There are forms of work other than pdV or displacement work. The following are the additional types of work transfer which may get involved in system-surroundings interactions.

(a) Electrical Work When a current flows through a resistor (Fig. 3.13), taken as a system, there is work transfer into the system. This is because the current can drive a motor, the motor can drive a pulley and the pulley can raise a weight.

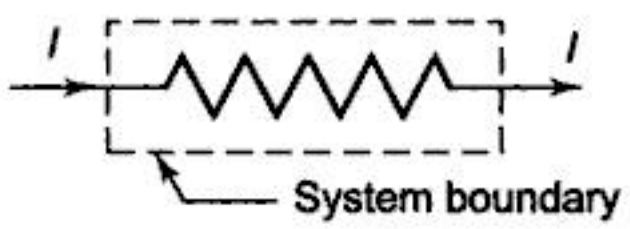

Fig. 3.13 *Electrical Work*

The current flow, I, in amperes, is given by

$$I = \frac{dC}{d\tau}$$

where C is the charge in coulombs and τ is time in seconds. Thus dC is the charge crossing a boundary during time $d\tau$. If E is the voltage potential, the work is

$$\delta W = E \cdot dC$$
$$= EI\, d\tau$$

$$\therefore \qquad W = \int_1^2 EI\, d\tau \qquad (3.12)$$

The electrical power will be

$$\dot{W} = \lim_{d\tau \to 0} \frac{dW}{d\tau} = EI \qquad (3.13)$$

This is the rate at which work is transferred.

(b) Shaft Work When a shaft, taken as the system (Fig. 3.14), is rotated by a motor, there is work transfer into the system. This is because the shaft can rotate a pulley which can raise a weight. If T is the torque applied to the shaft and $d\theta$ is the angular displacement of the shaft, the shaft work is

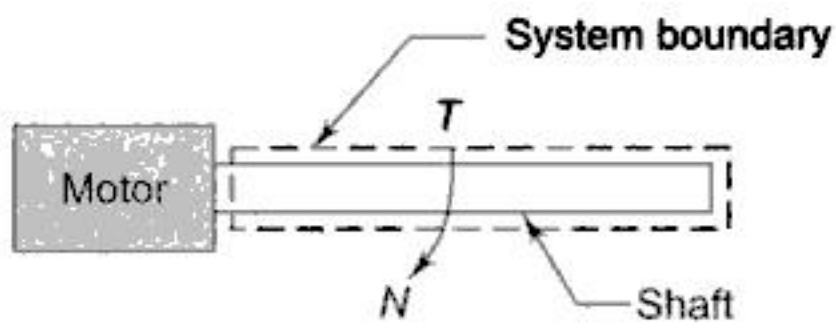

Fig. 3.14 *Shaft Work*

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$$W = \int_1^2 T d\theta \tag{3.14a}$$

and the shaft power is

$$\dot{W} = \int_1^2 T \frac{d\theta}{d\tau} = T\omega \tag{3.15b}$$

where ω is the angular velocity and T is considered a constant in this case.

(c) Paddle-Wheel Work or Stirring Work As the weight is lowered, and the paddle wheel turns (Fig. 3.15), there is work transfer into the fluid system which gets stirred. Since the volume of the system remains constant, $\int pdV = 0$. If m is the mass of the weight lowered through a distance dz and T is the torque transmitted by the shaft in rotating through an angle $d\theta$, the differential work transfer to the fluid is given by

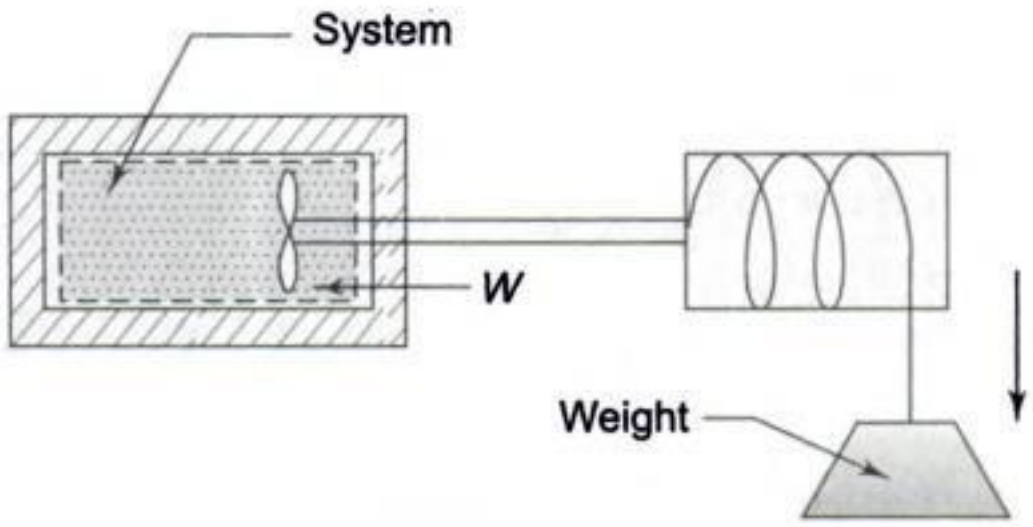

Fig. 3.15 *Paddle-Wheel Work*

$$dW = mgdz = Td\theta$$

and the total work transfer is

$$W = \int_1^2 mgdz = \int_1^2 W' \, dz = \int_1^2 Td\theta \tag{3.15}$$

where W' is the weight lowered.

(d) Flow Work The flow work, significant only in a flow process or an open system, represents the energy transferred across the system boundary as a result of the energy imparted to the fluid by a pump, blower or compressor to make the fluid flow across the control volume. Flow work is analogous to displacement work. Let p be the fluid pressure in the plane of the imaginary piston, which acts in a direction normal to it (Fig. 3.16). The work done on this imaginary piston by the external pressure as the piston moves forward is given by

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$$đ W_{\text{flow}} = p\,dV, \qquad (3.16)$$

where dV is the volume of fluid element about to enter the system.

$$\therefore \qquad đ W_{\text{flow}} = pv\,dm \qquad (3.17)$$

where $dV = v\,dm$

Therefore, flow work at inlet (Fig. 3.16),

$$(dW_{\text{flow}})_{\text{in}} = p_1 v_1\,dm_1 \qquad (3.18)$$

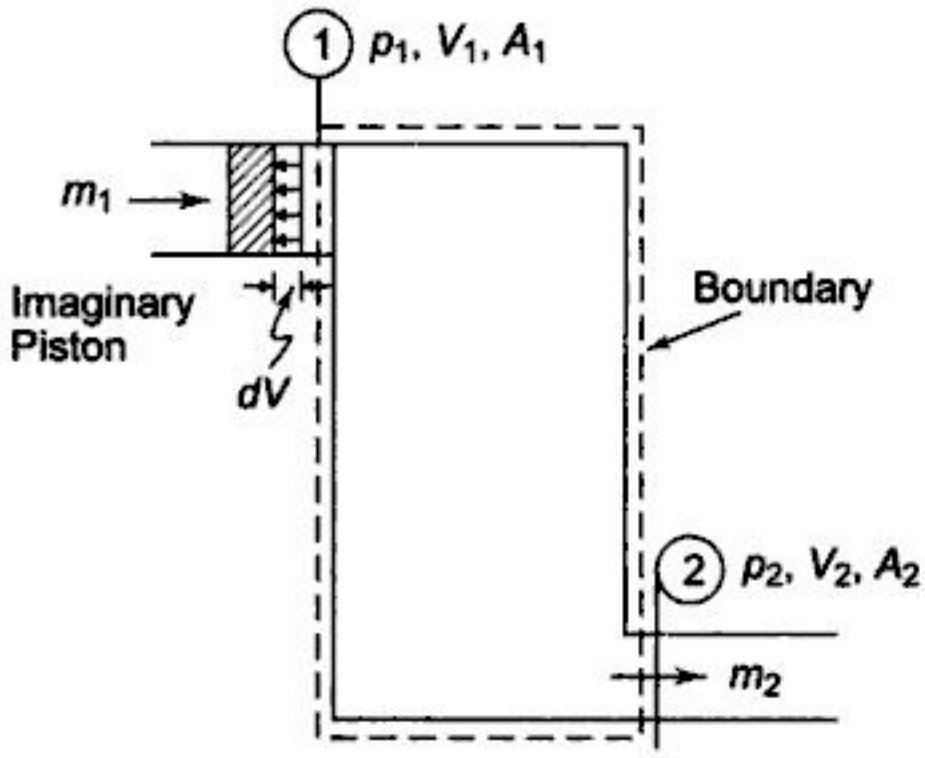

Fig. 3.16 Flow Work

Equation (3.18) can also be derived in a slightly different manner. If the normal pressure p_1 is exerted against the area A_1, giving a total force $(p_1 A_1)$ against the piston, in time $d\tau$, this force moves a distance $\mathbf{V}_1 d\tau$, where $\mathbf{V}_1$ is the velocity of flow (piston). The work in the time $d\tau$ is $p_1 A_1 \mathbf{V}_1 d\tau$, or the work per unit time is $p_1 A_1 \mathbf{V}_1$. Since the flow rate

$$\omega_1 = \frac{A_1 \mathbf{V}_1}{v_1} = \frac{dm_1}{d\tau}$$

the work done in time $d\tau$ becomes

$$(đ W_{\text{flow}})_{\text{in}} = p_1 v_1\,dm_1$$

Similarly, flow work of the fluid element leaving the system is

$$(đ W_{\text{flow}})_{\text{out}} = p_2 v_2\,dm_2 \qquad (3.19)$$

The flow work per unit mass is thus

$$đ W_{\text{flow}} = pv \qquad (3.20)$$

It is the displacement work done at the moving system boundary.

(e) Work Done in Stretching a Wire Let us consider a wire as the system. If the length of the wire in which there is a tension $\mathcal{F}$ is changed from L to $L + dL$, the infinitesimal amount of work that is done is equal to

$$đ W = -\mathcal{F}\,dL$$

The minus sign is used because a positive value of dL means an expansion of the wire, for which work must be done on the wire, i.e. negative work. For a finite change of length,

$$W = -\int_1^2 \mathcal{F}\,dL \qquad (3.21)$$

If we limit the problem to within the elastic limit, where E is the modulus of elasticity, s is the stress, ε is the strain, and A is the cross-sectional area, then

$$\mathcal{F} = sA = E\varepsilon A, \text{ since } \frac{s}{\varepsilon} = E$$

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$$d\varepsilon = \frac{dL}{L}$$

$$đ W = -\mathcal{F}\, dL = -E\varepsilon\, AL\, d\varepsilon$$

$$\therefore \qquad W = -A\varepsilon L \int_1^2 \varepsilon d\varepsilon = -\frac{AEL}{2}\,(\varepsilon_2^2 - \varepsilon_1^2) \qquad (3.22)$$

(f) Work Done in Changing the Area of a Surface Film A film on the surface of a liquid has a surface tension, which is a property of the liquid and the surroundings. The surface tension acts to make the surface area of the liquid a minimum. It has the unit of force per unit length. The work done on a homogeneous liquid film in changing its surface area by an infinitesimal amount dA is

$$đ W = -\sigma\, d\, A$$

where σ is the surface tension (N/m).

$$\therefore \qquad W = -\int_1^2 \sigma\, d\, A \qquad (3.23)$$

(g) Magnetization of a Paramagnetic Solid The work done per unit volume on a magnetic material through which the magnetic and magnetization fields are uniform is

$$đ W = -HdI$$

and

$$W_{1\text{–}2} = -\int_{I_1}^{I_2} HdI \qquad (3.24)$$

where H is the field strength, and I is the component of the magnetization field in the direction of the field. The minus sign provides that an increase in magnetization (positive dI) involves negative work.

The following equations summarize the different forms of work transfer:

Displacement work (compressible fluid)	$W = \int_1^2 pdV$
Electrical work	$W = \int_1^2 E\, dC = \int_1^2 EI\, d\tau$
Shaft work	$W = \int_1^2 T\, d\theta$
Surface film	$W = -\int_1^2 \sigma\, d\, A$ (3.25)

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Stretched wire $$W = -\int_1^2 \mathcal{F}\, dL$$

Magnetised solid $$W = -\int_1^2 H\, dI$$

It may be noted in the above expressions that the work is equal to the integral of the product of an intensive property and the change in its related extensive property. These expressions are valid only for infinitesimally slow quasi-static processes.

There are some other forms of work which can be identified in processes that are not quasi-static, for example, the work done by shearing forces in a process involving friction in a viscous fluid.

3.5 FREE EXPANSION WITH ZERO WORK TRANSFER

Work transfer is identified only at the boundaries of a system. It is a boundary phenomenon, and a form of energy in transit crossing the boundary. Let us consider a gas separated from the vacuum by a partition (Fig. 3.17). Let the partition be removed. The gas rushes to fill the entire volume. The expansion of a gas against vacuum is called *free expansion*. If we neglect the work associated with the removal of partition, and consider the gas and vacuum together as our system (Fig. 3.17a), there is no work transfer involved here, since no work crosses the system boundary, and hence

$$\int_1^2 đ W = 0, \quad \text{although} \quad \int_1^2 p dV \neq 0$$

If only the gas is taken as the system (Fig. 3.17b), when the partition is removed there is a change in the volume of the gas, and one is tempted to calculate the work from the expression $\int_1^2 p dV$. However, this is not a quasi-static process, although the initial and final end states are in equilibrium. Therefore, the work cannot be calculated from this relation. The two end states can be located on the p–V diagram and these are joined by a dotted line (Fig. 3.17c) to indicate that the process had occurred. However, if the vacuum space is divided into a large number of small volumes by partitions and the partitions are removed one by one slowly (Fig. 3.17d), then every state passed through by the system is an equilibrium state and the work done can then be estimated from the relation $\int_1^2 p dV$ (Fig. 3.17e).

Yet, in free expansion of a gas, thee is no resistance to the fluid at the system boundary as the volume of the gas increases to fill up the vacuum space. Work is done by a system to overcome some resistance. Since vacuum does not offer any resistance, there is no work transfer involved in free expansion.

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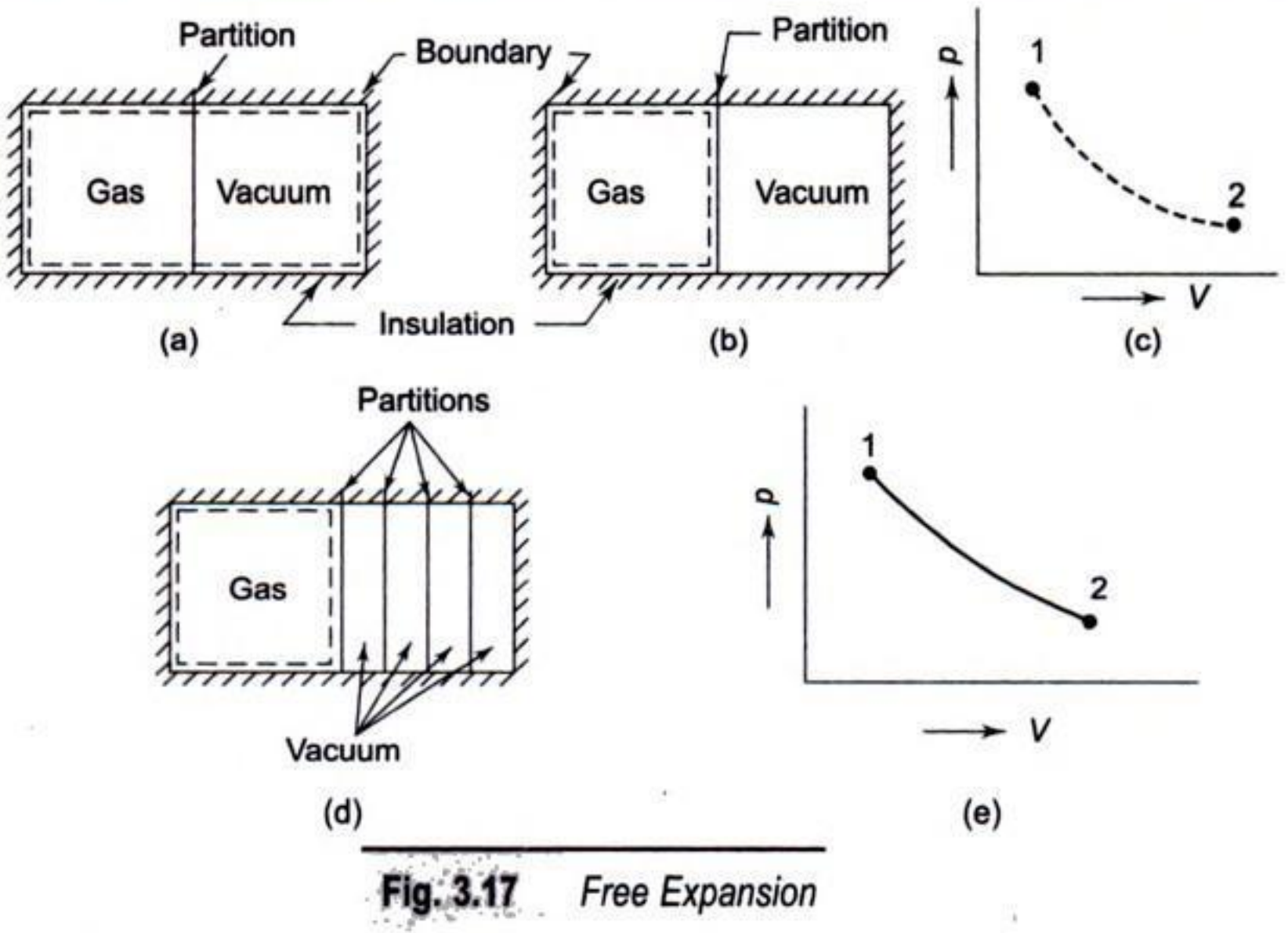

Fig. 3.17 *Free Expansion*

3.6 NET WORK DONE BY A SYSTEM

Often different forms of work transfer occur simultaneously during a process executed by a system. When all these work interactions have been evaluated, the total or net work done by the system would be equal to the algebraic sum of these as given below

$$W_{\text{total}} = W_{\text{displacement}} + W_{\text{shear}} + W_{\text{electrical}} + W_{\text{stirring}} + \cdots$$

3.7 HEAT TRANSFER

Heat is defined as the form of energy that is transferred across a boundary *by virtue of a temperature difference*. The temperature difference is the 'potential' or 'force' and heat transfer is the 'flux'.

The transfer of heat between two bodies in direct contact is called *conduction*. Heat may be transferred between two bodies separated by empty space or gases by the mechanism of *radiation* through electromagnetic waves. A third method of heat transfer is *convection* which refers to the transfer of heat between a wall and a fluid system in motion.

The direction of heat transfer is taken from the high temperature system to the low temperature system. *Heat flow into a system is taken to be positive, and heat flow out of a system is taken as negative* (Fig. 3.18). The symbol Q is used for heat transfer, i.e. the quantity of heat transferred within a certain time.

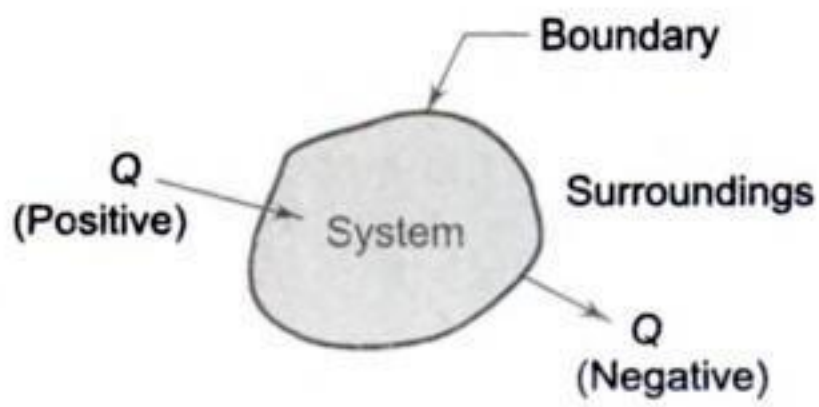

Fig. 3.18 *Direction of Heat Transfer*

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Heat is a form of energy in transit (like work transfer). It is a boundary phenomenon, since it occurs only at the boundary of a system. Energy transfer by virtue of temperature difference only is called heat transfer. All other energy interactions may be termed as work transfer.

Heat is not that which inevitably causes a temperature rise. When heat is transferred to an ice-and-water mixture, the temperature does not rise until all the ice has melted. When a temperature rise in a system occurs, it may not be due to heat transfer, since a temperature rise may be caused by work transfer also. Heat, like work, is not a conserved quantity, and is not a property of a system.

A process in which no heat crosses the boundary of the system is called an *adiabatic process*.

Thus, an adiabatic process is one in which there is only work interaction between the system and its surroundings.

A wall which is impermeable to the flow of heat is an *adiabatic wall*, whereas a wall which permits the flow of heat is a *diathermic wall*.

The unit of heat is Joule in S.I. units.

The rate of heat transfer or work transfer is given in kW or W.

3.8 HEAT TRANSFER—A PATH FUNCTION

Heat transfer is a *path function*, that is, the amount of heat transferred when a system changes from state 1 to state 2 depends on the intermediate states through which the system passes, i.e. its path. Therefore dQ is an inexact differential, and we write

$$\int_1^2 đQ = Q_{1\text{-}2} \quad \text{or} \quad {}_1Q_2$$

The displacement work is given by

$$W_{1\text{-}2} = \int_1^2 đW = \int_1^2 p dV$$

It is valid for a quasi-static process, and the work transfer involved is represented by the area under the path on p-v diagram (Fig. 3.19a). Whenever there is a difference in pressure, there will be displacement work. The pressure difference is the cause and work transfer is the effect. The work transfer is equal to the integral of the product of the intensive property, p and the differential change in the extensive property, dV.

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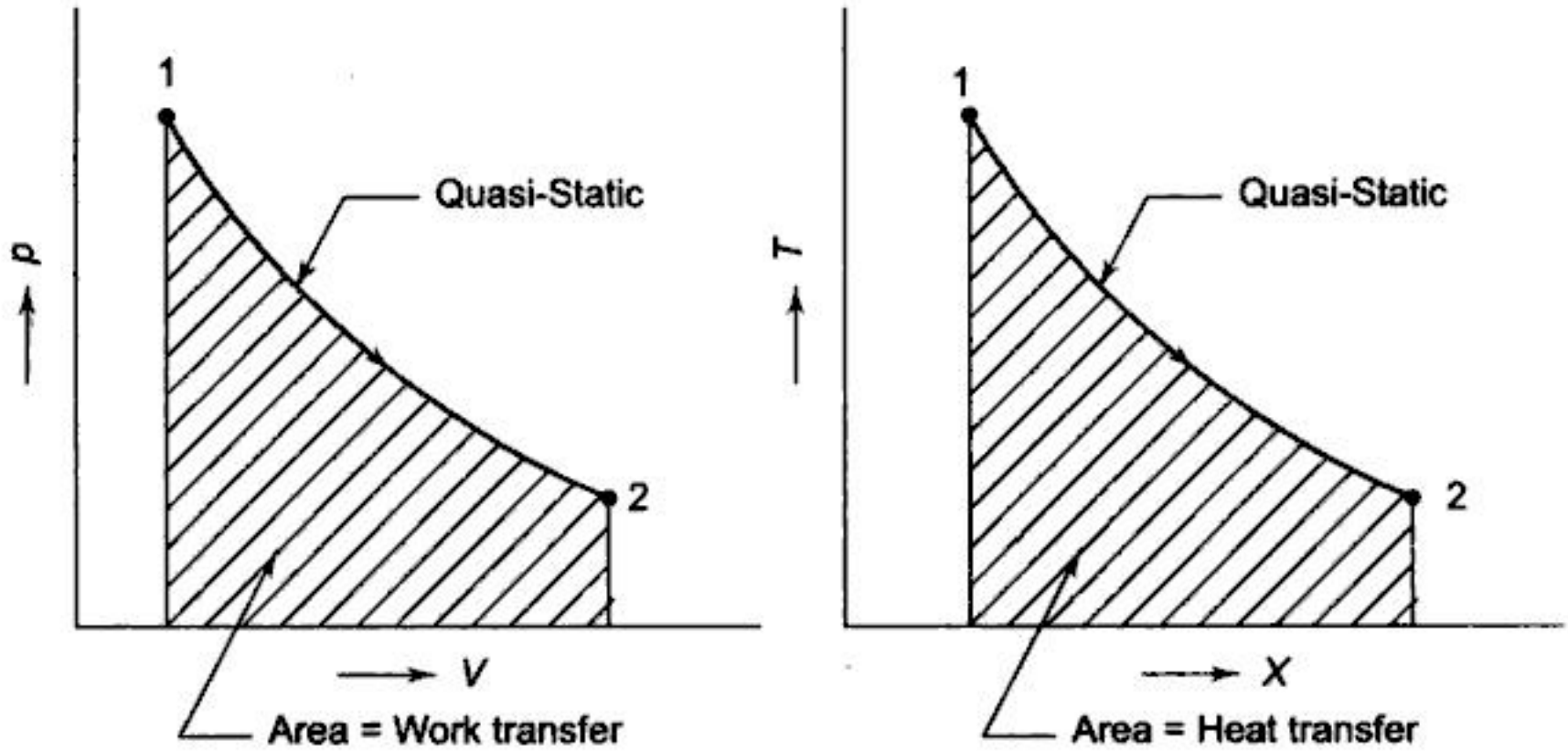

$$W_{1-2} = \int_1^2 p\,dV \quad \text{(a)} \qquad Q_{1-2} = \int_1^2 T\,dx \quad \text{(b)}$$

Fig. 3.19 *Representation of Work Transfer and Heat Transfer in Quasi-Static Processes on p-v and T-x Coordinates*

Likewise, whenever there is a difference in temperature, there will be heat flow. The temperature difference is the cause and heat transfer is the effect. Just like displacement work, the heat transfer can also be written as the integral of the product of the intensive property T and the differential change of an extensive property, say X (Fig. 3.19b).

$$Q_{1-2} = \int_1^2 đ Q = \int_1^2 T dX \tag{3.26}$$

It must also be valid for a quasi-static process only, and the heat transfer involved is represented by the area under the path 1–2 in T-X plot (Fig. 3.19b). Heat transfer is, therefore, a path function, i.e. the amount of heat transferred when a system changes from state 1 to state 2 depends on the path the system follows (Fig. 3.19b). Therefore $đ Q$ is an inexact differential. Now,

$$đ Q = T\,dX$$

where X is an extensive property and dX is an exact differential.

$$\therefore \qquad dX = \frac{1}{T} đ Q \tag{3.27}$$

To make $đ Q$ integrable, i.e. an exact differential, it must be multiplied by an integrating factor which is, in this case, $1/T$. The extensive property X is yet to be defined. It has been introduced in Chapter 7 and it is called 'entropy'.

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3.9 SPECIFIC HEAT AND LATENT HEAT

The *specific heat* of a substance is defined as the amount of heat required to raise a unit mass of the substance through a unit rise in temperature. The symbol c will be used for specific heat.

$$\therefore \qquad c = \frac{Q}{m \cdot \Delta t} \text{ J/kg K}$$

where Q is the amount of heat transfer (J), m, the mass of the substance (kg), and Δt, the rise in temperature (K).

Since heat is not a property, as explained later, so the specific heat is qualified with the process through which exchange of heat is made. For gases, if the process is at constant pressure, it is c_p, and if the process is at constant volume, it is c_v. For solids and liquids, however, the specific heat does not depend on the process. An elegant manner of defining specific heats, c_v and c_p, in terms of properties is given in Secs 4.5 and 4.6.

The product of mass and specific heat (mc) is called the *heat capacity* of the substance. The capital letter C, C_p or C_v, is used for heat capacity.

The latent heat is the amount of heat transfer required to cause a phase change in unit mass of a substance *at a constant pressure and temperature*. There are three phases in which matter can exist: solid, liquid, and vapour or gas. The *latent heat of fusion* (l_{fu}) is the amount of heat transferred to melt unit mass of solid into liquid, or to freeze unit mass of liquid to solid. The *latent heat of vaporization* (l_{vap}) is the quantity of heat required to vaporize unit mass of liquid into vapour, or condense unit mass of vapour into liquid. The *latent heat of sublimation* (l_{sub}) is the amount of heat transferred to convert unit mass of solid to vapour or vice versa. l_{fu} is not much affected by pressure, whereas l_{vap} is highly sensitive to pressure.

3.10 POINTS TO REMEMBER REGARDING HEAT TRANSFER AND WORK TRANSFER

(a) Heat transfer and work transfer are the *energy interactions*. A closed system and its surroundings can interact in two ways: by heat transfer and by work transfer. Thermodynamics studies how these interactions bring about property changes in a system.

(b) The same effect in a closed system can be brought about either by heat transfer or by work transfer. Whether heat transfer or work transfer has taken place depends on what constitutes the system.

(c) Both heat transfer and work transfer are boundary phenomena. Both are observed at the boundaries of the system, and both represent energy crossing the boundaries of the system.

(d) *It is wrong to say 'total heat' or 'heat content' of a closed system*, because heat or work is not a property of the system. Heat, like work, cannot be stored by the system. Both heat and work are the energy is transit.

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(e) Heat transfer is the energy interaction due to temperature difference only. All other energy interactions may be termed as work transfer.

(f) Both heat and work are path functions and inexact differentials. The magnitude of heat transfer or work transfer depends upon the path the system follows during the change of state.

Solved Examples

Example 3.1 Gas from a bottle of compressed helium is used to inflate an inelastic flexible balloon, originally folded completely flat to a volume of 0.5 m^3. If the barometer reads 760 mm Hg, what is the amount of work done upon the atmosphere by the balloon? Sketch the system before and after the process.

Solution The firm line P_1 (Fig. 3.20) shows the boundary of the system before the process, and the dotted line P_2 shows the boundary after the process. The displacement work

$$W_d = \int_{\text{Balloon}} pdV + \int_{\text{Bottle}} pdV = p\,\Delta V + 0$$

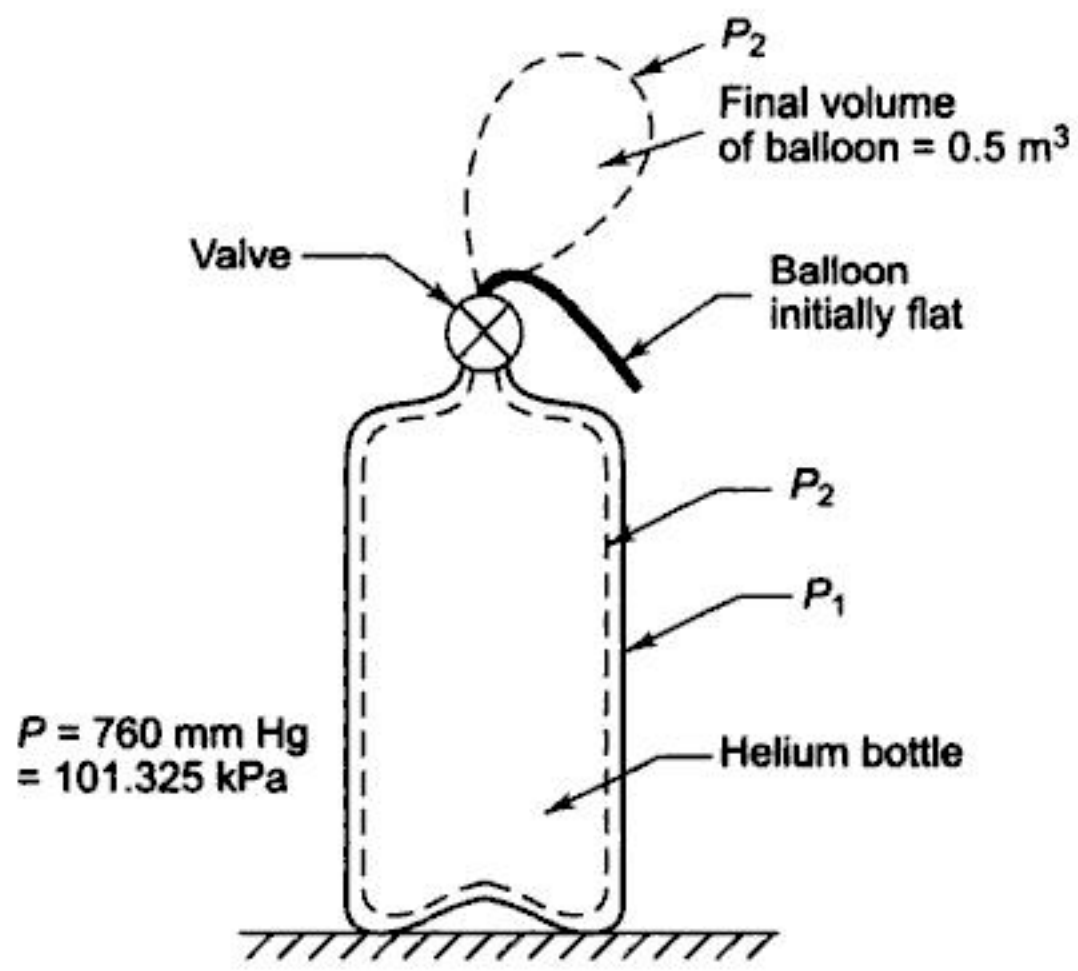

Fig. 3.20

$$= 101.325\ \frac{\text{kN}}{\text{m}^2} \times 0.5\ \text{m}^3 = \mathbf{50.66\ kJ}$$

This is positive, because work is done by the system. Work done by the atmosphere is –50.66 kJ. Since the wall of the bottle is rigid, there is no pdV-work involved in it.

It is assumed that the pressure in the balloon is atmospheric at all times, since the balloon fabric is light, inelastic and unstressed. If the balloon were elastic and

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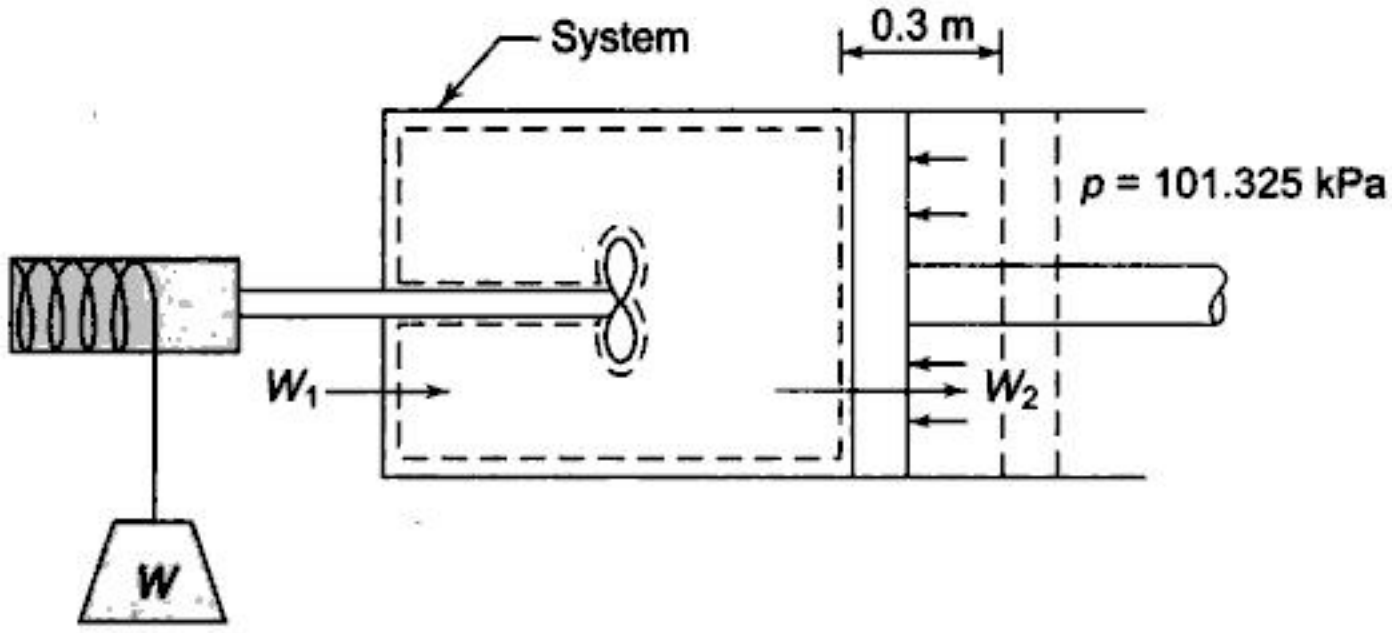

Fig. 3.22

Solution Work done by the stirring device upon the system (Fig. 3.22)

$$W_1 = 2\pi TN$$

$$= 2\pi \times 1.275 \times 10{,}000 \text{ Nm} = \mathbf{80 \text{ kJ}}$$

This is negative work for the system.

Work done by the system upon the surroundings

$$W_2 = (pA) \cdot L$$

$$= 101.325 \frac{\text{kN}}{\text{m}^2} \times \frac{\pi}{4} (0.6)^2 \text{ m}^2 \times 0.80 \text{ m} = 22.9 \text{ kJ}$$

This is positive work for the system. Hence, the net work transfer for the system

$$W = W_1 + W_2 = -80 + 22.9 = \mathbf{-57.1 \text{ kJ}}$$

Example 3.4 The following data refer to a 12-cylinder, single-acting, two-stroke marine diesel engine:

Speed—150 rpm

Cylinder diameter—0.8 m

Stroke of piston—1.2 m

Area of indicator diagram—5.5×10^{-4} m^2

Length of diagram—0.06 m

Spring value—147 MPa per m

Find the net rate of work transfer from the gas to the pistons in kW.

Solution Mean effective pressure, p_m, is given by

$$p_m = \frac{a_d}{l_d} \times \text{spring constant}$$

$$= \frac{5.5 \times 10^{-4} \text{ m}^2}{0.06} \times 147 \frac{\text{MPa}}{\text{m}} = 1.35 \text{ MPa}$$

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3.5 A mass of gas is compressed in a quasi-static process from 80 kPa, 0.1 m^3 to 0.4 MPa, 0.03 m^3. Assuming that the pressure and volume are related by pv^n = constant, find the work done by the gas system.

Ans. −11.83 kJ

3.6 A single-cylinder, double-acting, reciprocating water pump has an indicator diagram which is a rectangle 0.075 m long and 0.05 m high. The indicator spring constant is 147 MPa per m. The pump runs at 50 rpm. The pump cylinder diameter is 0.15 m and the piston stroke is 0.20 m. Find the rate in kW at which the piston does work on the water.

Ans. 43.3 kW

3.7 A single-cylinder, single-acting, 4 stroke engine of 0.15 m bore develops an indicated power of 4 kW when running at 216 rpm. Calculate the area of the indicator diagram that would be obtained with an indicator having a spring constant of 25×10^6 N/m^3. The length of the indicator diagram is 0.1 times the length of the stroke of the engine.

Ans. 505 mm^2

3.8 A six-cylinder, 4-stroke gasoline engine is run at a speed of 2520 RPM. The area of the indicator card of one cylinder is 2.45×10^3 mm^2 and its length is 58.5 mm. The spring constant is 20×10^6 N/m^3. The bore of the cylinders is 140 mm and the piston stroke is 150 mm. Determine the indicated power, assuming that each cylinder contributes an equal power.

Ans. 243.57 kW

3.9 A closed cylinder of 0.25 m diameter is fitted with a light frictionless piston. The piston is retained in position by a catch in the cylinder wall and the volume on one side of the piston contains air at a pressure of 750 kN/m^2. The volume on the other side of the piston is evacuated. A helical spring is mounted coaxially with the cylinder in this evacuated space to give a force of 120 N on the piston in this position. The catch is released and the piston travels along the cylinder until it comes to rest after a stroke of 1.2 m. The piston is then held in its position of maximum travel by a ratchet mechanism. The spring force increases linearly with the piston displacement to a final value of 5 kN. Calculate the work done by the compressed air on the piston.

Ans. 3.07 kJ

3.10 A steam turbine drives a ship's propeller through an 8 : 1 reduction gear. The average resisting torque imposed by the water on the propeller is 750×10^3 mN and the shaft power delivered by the turbine to the reduction gear is 15 MW. The turbine speed is 1450 rpm. Determine (a) the torque developed by the turbine, (b) the power delivered to the propeller shaft, and (c) the net rate of working of the reduction gear.

Ans. (a) T = 98.84 km N, (b) 14.235 MW, (c) 0.765 MW

3.11 A fluid, contained in a horizontal cylinder fitted with a frictionless leakproof piston, is continuously agitated by means of a stirrer passing through the cylinder cover. The cylinder diameter is 0.40 m. During the stirring process lasting 10 minutes, the piston slowly moves out a distance of 0.485 m against the

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Prior to Joule, heat was considered to be an invisible fluid flowing from a body of higher calorie to a body of lower calorie, and this was known as the *caloric theory of heat*. It was Joule who first established that heat is a form of energy, and thus laid the foundation of the first law of thermodynamics.

4.2 FIRST LAW FOR A CLOSED SYSTEM UNDERGOING A CHANGE OF STATE

The expression $(\Sigma W)_{cycle} = (\Sigma Q)_{cycle}$ applies only to systems undergoing cycles, and the algebraic summation of all energy transfer across system boundaries is zero. But if a system undergoes a change of state during which both heat transfer and work transfer are involved, the *net* energy transfer will be stored or accumulated within the system. If Q is the amount of heat transferred to the system and W is the amount of work transferred from the system during the process (Fig. 4.3), the net energy transfer $(Q - W)$ will be stored in the system. Energy in storage is neither heat nor work, and is given the name *internal energy* or simply, the *energy* of the system.

Therefore $$Q - W = \Delta E$$

where ΔE is the increase in the energy of the system

or $$Q = \Delta E + W \qquad (4.2)$$

Here Q, W, and ΔE are all expressed in the same units (in joules). Energy may be stored by a system in different modes, as explained in Article 4.4.

If there are more energy transfer quantities involved in the process, as shown in Fig. 4.4, the first law gives

$$(Q_2 + Q_3 - Q_1) = \Delta E + (W_2 + W_3 - W_1 - W_4)$$

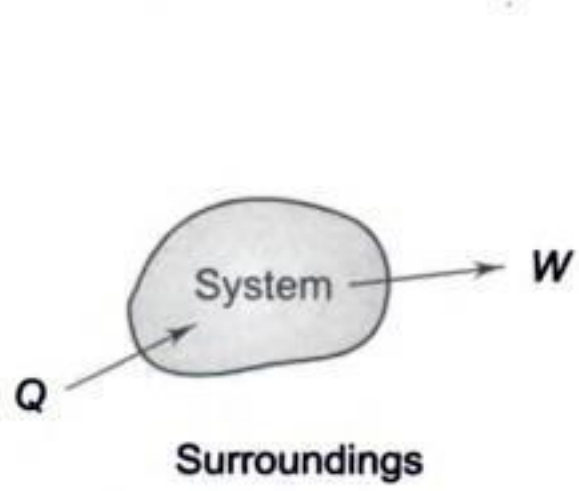

Fig. 4.3 *Heat and Work Interactions of a System with Its Surroundings in a Process*

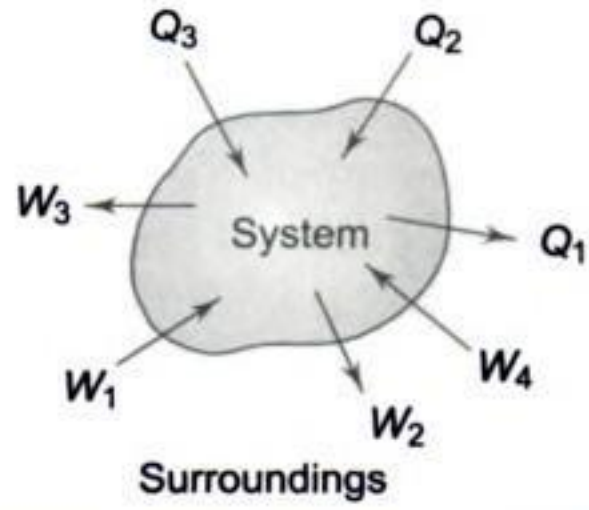

Fig. 4.4 *System-surroundings Interaction in a Process Involving Many Energy Fluxes*

Energy is thus conserved in the operation. The first law is a particular formulation of the principle of the conservation of energy. Equation (4.2) may also be considered as the definition of energy. This definition does not give an absolute value of energy E, but only the change of energy ΔE for the process. It can, however, be shown that the energy has a definite value at every state of a system and is, therefore, a property of the system.

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where $đW = đW_{pdV} + đW_{\text{shaft}} + đW_{\text{electrical}} + \cdots,$

considering the different forms of work transfer which may be present. When only pdV work is present, the equations become

$$đQ = dE + pdV \tag{4.16}$$

$$đQ = dU + pdV \tag{4.17}$$

or, in the integral form

$$Q = \Delta E + \int pdV \tag{4.18}$$

$$Q = \Delta U + \int pdV \tag{4.19}$$

4.5 SPECIFIC HEAT AT CONSTANT VOLUME

The specific heat of a substance at constant volume c_V is defined as the rate of change of specific internal energy with respect to temperature when the volume is held constant, i.e.

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v \tag{4.20}$$

For a constant-volume process

$$(\Delta u)_v = \int_{T_1}^{T_2} c_v \cdot dT \tag{4.21}$$

The first law may be written for a closed stationary system composed of a unit mass of a pure substance

$$Q = \Delta u + W$$

or $$đQ = du + đW$$

For a process in the absence of work other than pdV work

$$đW = pdV$$

$$\therefore \quad đQ = dU + pdV \tag{4.22}$$

When the volume is held constant

$$(Q)_v = (\Delta u)_v$$

$$\therefore \quad (Q)_v = \int_{T_1}^{T_2} c_v \, dT \tag{4.23}$$

Heat transferred at constant volume increases the internal energy of the system. If the specific heat of a substance is defined in terms of heat transfer, then

$$c_v = \left(\frac{\partial Q}{\partial T}\right)_u$$

Since Q is not a property, this definition does not imply that c_v is a property of a substance. Therefore, this is not the appropriate method of defining the specific heat, although $(đQ)_v = du$.

Since u, T, and v are properties, c_v is a property of the system. The product $mc_v = C_v$ is called the *heat capacity at constant volume* (J/K).

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The system liberates 73 kJ of heat

(c) $$W_{adb} = W_{ad} + W_{db} = W_{ad} = 10.5 \text{ kJ}$$

$\therefore$ $$Q_{ad} = U_d - U_a + W_{ad}$$
$$= 42 - 0 + 10.5 = 52.5 \text{ kJ}$$

Now $$Q_{adb} = 62.5 \text{ kJ} = Q_{ad} + Q_{db}$$

$\therefore$ $$Q_{db} = 62.5 - 52.5 = \mathbf{10 \text{ kJ}}$$

Example 4.3 A piston and cylinder machine contains a fluid system which passes through a complete cycle of four processes. During a cycle, the sum of all heat transfers is – 170 kJ. The system completes 100 cycles per min. Complete the following table showing the method for each item, and compute the net rate of work output in kW.

Process	Q (kJ/min)	W (kJ/min)	ΔE (kJ/min)
a–b	0	2,170	—
b–c	21,000	0	—
c–d	–2,100	—	–36,600
d–a	—	—	—

Solution Process a–b:

$$Q = \Delta E + W$$
$$0 = \Delta E + 2170$$

$\therefore$ $$\Delta E = -2170 \text{ kJ/min}$$

Process b–c:

$$Q = \Delta E + W$$
$$21{,}000 = \Delta E + 0$$

$\therefore$ $$\Delta E = 21{,}000 \text{ kJ/min}$$

Process c–d:

$$Q = \Delta E + W$$
$$-2100 = -36{,}600 + W$$

$\therefore$ $$W = 34{,}500 \text{ kJ/min}$$

Process d–a:

$$\sum_{\text{cycle}} Q = -170 \text{ kJ}$$

The system completes 100 cycles/min.

$\because$ $$Q_{ab} + Q_{bc} + Q_{cd} + Q_{da} = -17{,}000 \text{ kJ/min}$$
$$0 + 21{,}000 - 2100 + Q_{da} = -17{,}000$$

$\therefore$ $$Q_{da} = -35{,}900 \text{ kJ/min}$$

Now $\oint dE = 0$, since cyclic integral of any property is zero.

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This cycle is known as *Lenoir cycle*, and the operation of pulse jet approximates to it.

Review Questions

4.1 State the first law for a closed system undergoing a cycle.
4.2 What was the contribution of J.P. Joule in establishing the first law?
4.3 What is the caloric theory of heat? Why was it rejected?
4.4 Which is the property introduced by the first law?
4.5 State the first law for a closed system undergoing a change of state.
4.6 Show that energy is a property of a system.
4.7 What are the modes in which energy is stored in a system?
4.8 Define internal energy. How is energy stored in molecules and atoms?
4.9 What is the difference between the standard symbols of E and U?
4.10 What is the difference between heat and internal energy?
4.11 Define enthalpy. Why does the enthalpy of an ideal gas depend only on temperature?
4.12 Define the specific heats at constant volume and at constant pressure.
4.13 Why should specific heat not be defined in terms of heat transfer?
4.14 Which property of a system increases when heat is transferred: (a) at constant volume, (b) at constant pressure?
4.15 What is a PMM1? Why is it impossible?

Problems

4.1 An engine is tested by means of a water brake at 1000 rpm. The measured torque of the engine is 10000 mN and the water consumption of the brake is 0.5 m^3/s, its inlet temperature being 20°C. Calculate the water temperature at exit, assuming that the whole of the engine power is ultimately transformed into heat which is absorbed by the cooling water. Ans. 20.5°C

4.2 In a cyclic process, heat transfers are + 14.7 kJ, – 25.2 kJ, – 3.56 kJ and + 31.5 kJ. What is the net work for this cyclic process? Ans. 17.34 kJ

4.3 A slow chemical reaction takes place in a fluid at the constant pressure of 0.1 MPa. The fluid is surrounded by a perfect heat insulator during the reaction which begins at state 1 and ends at state 2. The insulation is then removed and 105 kJ of heat flow to the surroundings as the fluid goes to state 3. The following data are observed for the fluid at states 1, 2 and 3.

State	$V(m^3)$	$t(°C)$
1	0.003	20
2	0.3	370
3	0.06	20

For the fluid system, calculate E_2 and E_3, if $E_1 = 0$

Ans. $E_2 = -29.7$ kJ, $E_3 = -110.7$ kJ

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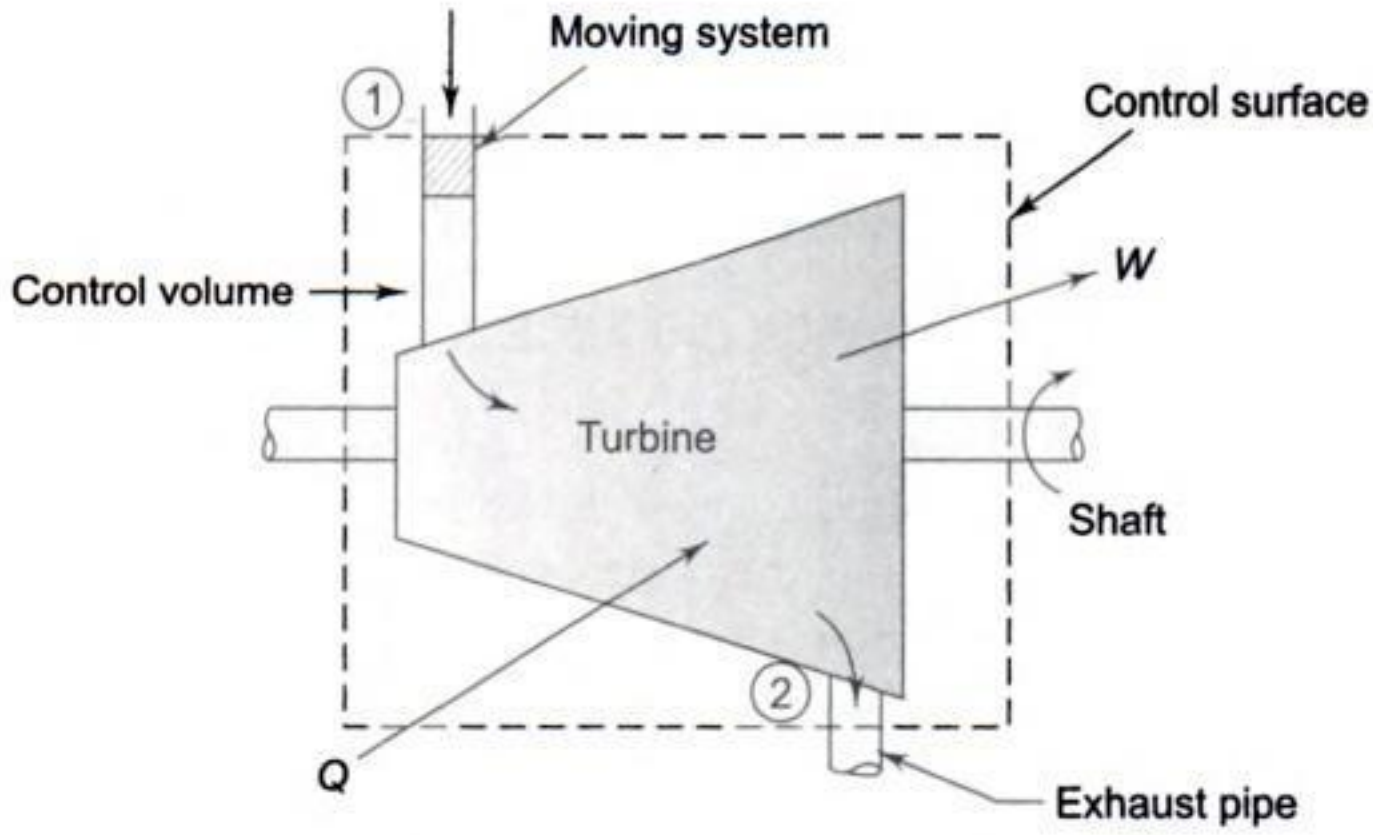

Fig. 5.1 *Flow Process Involving Work and Heat Interactions*

Although the system approach is quite valid, there is another approach which is found to be highly convenient. Instead of concentrating attention upon a certain quantity of fluid, which constitutes a moving system in flow processes, attention is focused upon a certain fixed region in space called a *control volume* through which the moving substance flows. This is similar to the analysis of Euler in fluid mechanics.

To distinguish the two concepts, it may be noted that while the system (closed) boundary usually changes shape, position and orientation relative to the observer, the control volume boundary remains fixed and unaltered. Again, while matter usually crosses the control volume boundary, no such flow occurs across the system boundary.

The broken line in Fig. 5.1 represents the surface of the control volume which is known as the *control surface*. This is the same as the system boundary of the open system. The method of analysis is to inspect the control surface and account for all energy quantities transferred through this surface. Since there is mass transfer across the control surface, a mass balance also has to be made. Sections 1 and 2 allow mass transfer to take place, and Q and W are the heat and work interactions respectively.

5.2 STEADY FLOW PROCESS

As a fluid flows through a certain control volume, its thermodynamic properties may vary along the space coordinates as well as with time. If the rates of flow of mass and energy through the control surface change with time, the mass and energy within the control volume also would change with time.

'Steady flow' means that the rates of flow of mass and energy across the control surface are constant.

In most engineering devices, there is a constant rate of flow of mass and energy through the control surface, and the control volume in course of time attains a

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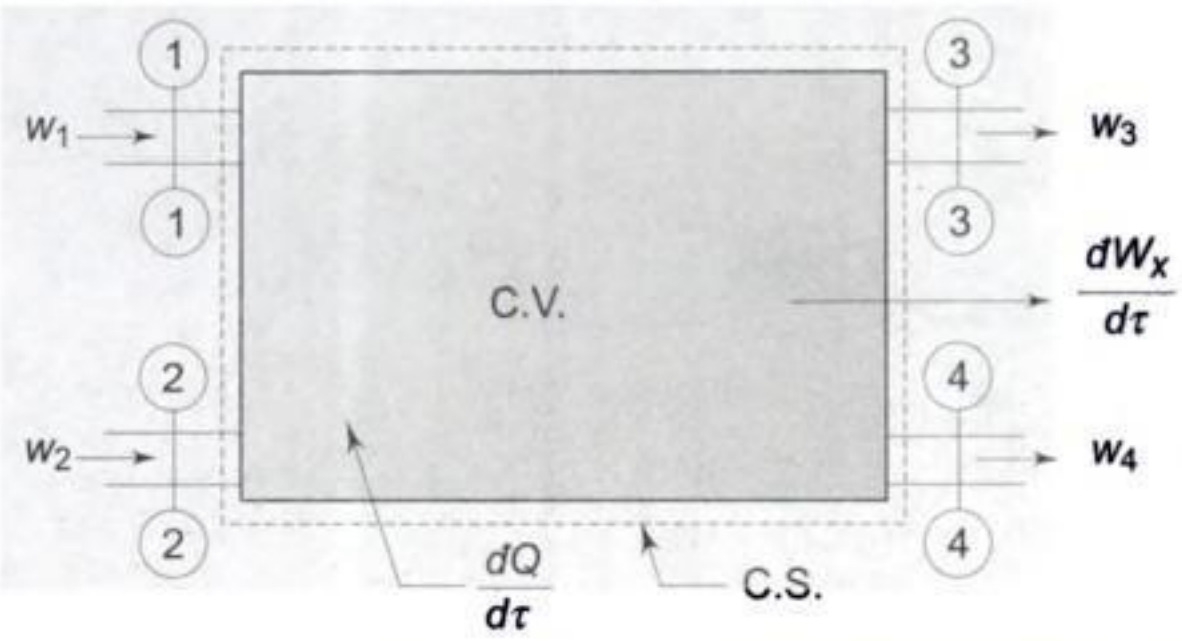

Fig. 5.3 *Steady Flow Process Involving Two Fluid Streams at the Inlet and Exit of the Control Volume*

Mass balance

$$w_1 + w_2 = w_3 + w_4 \tag{5.11}$$

$$\frac{A_1\mathbf{V}_1}{v_1} + \frac{A_2\mathbf{V}_2}{v_2} = \frac{A_3\mathbf{V}_3}{v_3} + \frac{A_4\mathbf{V}_4}{v_4} \tag{5.12}$$

Energy balance

$$w_1\left(h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g\right) + w_2\left(h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g\right) + \frac{đQ}{d\tau}$$

$$= w_3\left(h_3 + \frac{\mathbf{V}_3^2}{2} + Z_3 g\right) + w_4\left(h_4 + \frac{\mathbf{V}_4^2}{2} + Z_4 g\right) + \frac{đW_x}{d\tau} \tag{5.13}$$

The steady flow energy equation applies to a wide variety of processes like pipe line flows, heat transfer processes, mechanical power generation in engines and turbines, combustion processes, and flows through nozzles and diffusors. In certain problems, some of the terms in steady flow energy equation may be negligible or zero. But it is best to write the full equation first, and then eliminate the terms which are unnecessary.

5.4 SOME EXAMPLES OF STEADY FLOW PROCESSES

The following examples illustrate the applications of the steady flow energy equation in some of the engineering systems.

5.4.1 Nozzle and Diffusor

A nozzle is a device which increases the velocity or K.E. of a fluid at the expense of its pressure drop, whereas a diffusor increases the pressure of a fluid at the expense of its K.E. Figure 5.4 shows a nozzle which is insulated. The steady flow energy equation of the control surface gives

$$h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g + \frac{đQ}{dm} = h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g + \frac{đW_x}{dm}$$

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5.5 COMPARISON OF S.F.E.E. WITH EULER AND BERNOULLI EQUATIONS

The steady flow energy equation (Eq. 5.8) can be written as

$$\frac{đQ}{dm} = (h_2 - h_1) + \frac{\overline{\mathbf{V}}_2^2 - \overline{\mathbf{V}}_1^2}{2} + (Z_2 - Z_1)\, g + \frac{dW_x}{dm}$$

In the differential form the S.F.E.E. becomes

$$đQ = dh + \overline{\mathbf{V}} d\overline{\mathbf{V}} + gdZ + dW_x \qquad (5.17)$$

where $đQ$ and $đW_x$ refer to unit mass of the substance. Since $h = u + pv$ and $đQ = đu + pdv$ (for a quasi-static path involving only pdv-work), Eq. (5.17) can be written as

$$du + p\mathbf{dv} = du + p\mathbf{dv} + \mathbf{v}dp + \mathbf{V}d\mathbf{V} + gdZ + đW_x$$

For an inviscid frictionless fluid flowing through a pipe

$$\mathbf{v}dp + \mathbf{V}d\mathbf{V} + gdZ = 0 \qquad (5.18)$$

This is the *Euler equation*. If we integrate between two Sections 1 and 2 of the pipe

$$\int_1^2 \mathbf{v}dp + \int_1^2 \mathbf{V}d\mathbf{V} + \int_1^2 gdZ = 0$$

For an incompressible fluid, v = constant

$$\therefore \qquad \mathbf{v}(p_2 - p_1) + \frac{\mathbf{V}_2^2}{2} - \frac{\mathbf{V}_1^2}{\mathbf{1}} + g\,(Z_2 - Z_1) = 0 \qquad (5.19)$$

Since the specific volume v is the reciprocal of the density ρ, we have

$$\frac{p_1}{\rho} + \frac{\mathbf{V}_1^2}{2} + Z_1 g = \frac{p_2}{\rho} + \frac{\mathbf{V}_2^2}{2} + Z_2 g \qquad (5.20)$$

or

$$\frac{p}{\rho} + \frac{\mathbf{V}^2}{2} + Zg = \text{constant} \qquad (5.21)$$

This is known as the *Bernoulli equation*, which is valid for an inviscid incompressible fluid. It can also be expressed in the following form

$$\Delta\left(p\mathbf{v} + \frac{\mathbf{V}^2}{2} + gZ\right) = 0 \qquad (5.22)$$

where v is constant and Δ (...) means 'increase in ...

The S.F.E.E. as given by Eq. (5.8) or Eq. (5.17) can be written with $(u + pv)$ substituted for h, as follows:

$$Q - W_x = \Delta\left(u + p\mathbf{v} + \frac{\mathbf{V}^2}{2} + gZ\right) \qquad (5.23)$$

A comparison of Eqs (5.22) and (5.23) shows that they have several terms in common. However, while the Bernoulli equation is restricted to frictionless in-

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$$= m_2u_2 - m_1u_1 - (m_2 - m_1)\left(\frac{\mathbf{V}_p^2}{2} + h_p\right) \tag{5.30}$$

which gives the energy balance for the process.

Control Volume Technique

Assume a control volume bounded by a control surface, as shown in Fig. 5.11. Applying the energy Eq. (5.27) to this case, the following energy balance may be written on a time rate basis

$$\frac{dE_V}{d\tau} = \frac{đQ}{d\tau} + \left(h_p + \frac{\mathbf{V}_p^2}{2}\right)\frac{dm}{d\tau}$$

Since h_p and $\mathbf{V}_p$ are constant, the equation is integrated to give for the total process

$$\Delta E_V = Q + \left(h_p + \frac{\mathbf{V}_p^2}{2}\right)(m_2 - m_1)$$

Now

$$\Delta E_V = U_2 - U_1 = m_2u_2 - m_1u_1$$

$$\therefore \qquad Q = m_2u_2 - m_1u_1 - \left(h_p + \frac{\mathbf{V}_p^2}{2}\right)(m_2 - m_1)$$

This equation is the same as Eq. (5.30).

5.8 DISCHARGING AND CHARGING A TANK

Let us consider a tank discharging a fluid into a supply line (Fig. 5.12). Since $đW_x = 0$ and $dm_{in} = 0$, applying first law to the control volume,

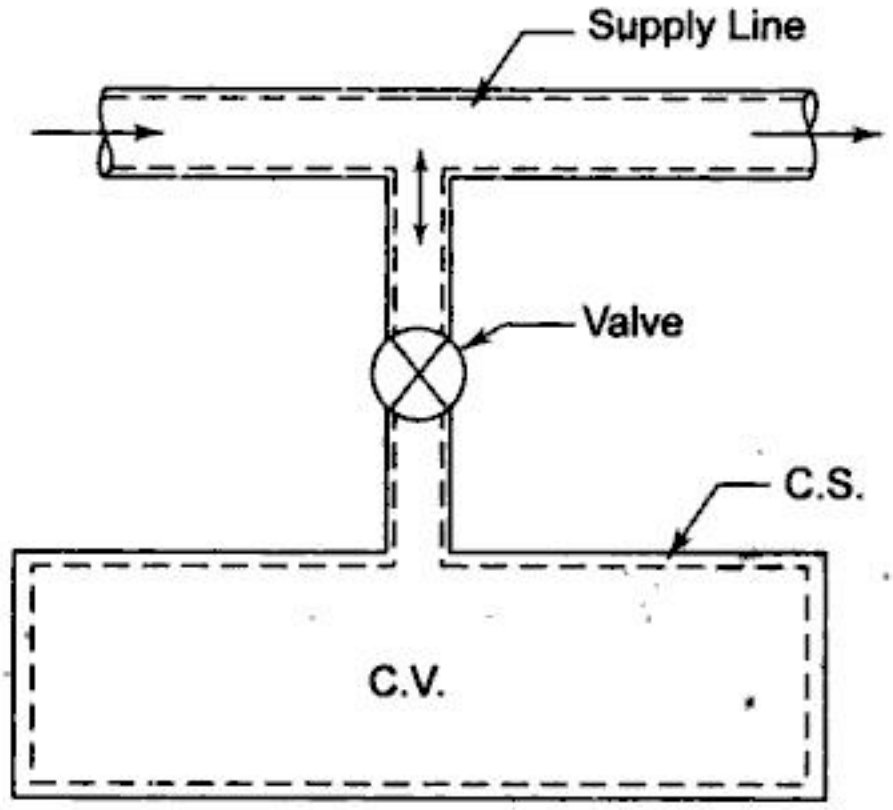

Fig. 5.12 *Charging and Discharging a Tank*

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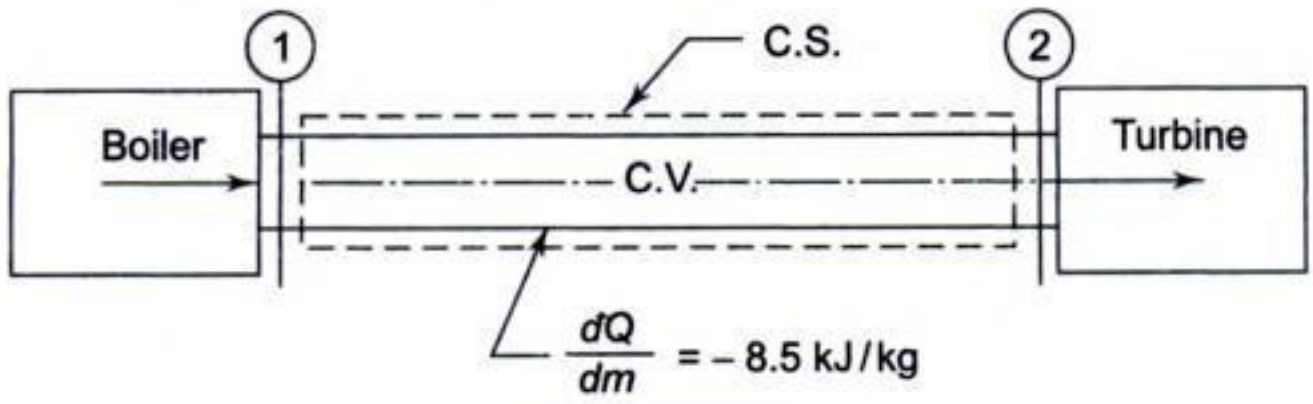

Fig. 5.15

Now
$$\frac{A_1\mathbf{V}_1}{v_1} = \frac{A_2\mathbf{V}_2}{v_2}$$

$$\therefore \quad \mathbf{V}_2 = \frac{A_1\mathbf{V}_1}{v_1}\cdot\frac{v_2}{A_2} = \frac{v_2}{v_1}\cdot\mathbf{V}_1 = \frac{0.084}{0.073}\,\mathbf{V}_1 = 1.15\,\mathbf{V}_1$$

and
$$\frac{đW_x}{dm} = 0$$

$$h_1 + \frac{\mathbf{V}_1^2}{2} + \frac{đQ}{dm} = h_2 + \frac{\mathbf{V}_2^2}{2}$$

$$\therefore \quad \frac{\left(\mathbf{V}_2^2 - \mathbf{V}_1^2\right)\times 10^{-3}}{2} = h_1 - h_2 + \frac{đQ}{dm}$$

$$= 3213.6 - 3202.6 + (-8.5) = \mathbf{2.5\ kJ/kg}$$

$$\mathbf{V}_1^2\,(1.15^2 - 1^2) = 5\times 10^3$$

$$\mathbf{V}_1^2 = 15{,}650\ \mathrm{m^2/s^2}$$

$$\therefore \quad \mathbf{V}_1 = 125.1\ \mathrm{m/s}$$

$$\therefore \quad \text{Mass flow rate } w = \frac{A_1\mathbf{V}_1}{v_1} = \frac{\frac{\pi}{4}\times(0.2)^2\,\mathrm{m^2}\times 125.1\,\mathrm{m/s}}{0.073\,\mathrm{m^3/kg}} = \mathbf{53.8\ kg/s}$$

Example 5.4 A certain water heater operates under steady flow conditions receiving 4.2 kg/s of water at 75°C temperature, enthalpy 313.93 kJ/kg. The water is heated by mixing with steam which is supplied to the heater at temperature 100.2°C and enthalpy 2676 kJ/kg. The mixture leaves the heater as liquid water at temperature 100°C and enthalpy 419 kJ/kg. How much steam must be supplied to the heater per hour?

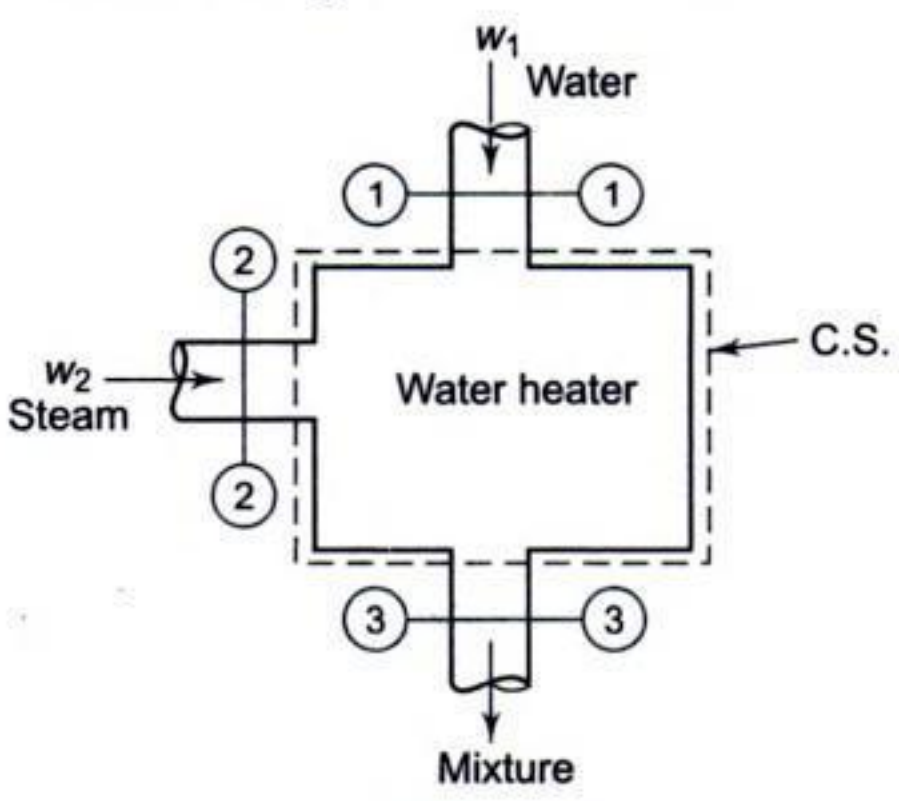

Fig. 5.16

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Solution Energy equation for the turbojet engine (Fig. 5.19) gives

$$w_a\left(h_a+\frac{\mathbf{V}_a^2}{2}\right)+w_f E_f+Q=w_g\left(h_g+\frac{\mathbf{V}_a^2}{2}+E_g\right)$$

$$\left(260+\frac{270^2\times10^{-3}}{2}\right)+0.0190\times44500-21$$

$$=1.0190\left(912+\frac{\mathbf{V}_g^2\times10^{-3}}{2}+0.05\frac{0.019}{1.019}\times44500\right)$$

$$260+36.5+845-21=1.019\left(912+\frac{\mathbf{V}_g^2\times10^{-3}}{2}+42\right)$$

$$\therefore\quad \frac{\mathbf{V}_g^2}{2}=156\times10^3\ \text{m}^2/\text{s}^2$$

$$\mathbf{V}_g=\sqrt{3.12}\times100\ \text{m/s}$$

Velocity of exhaust gas, $\mathbf{V}_g=560$ m/s

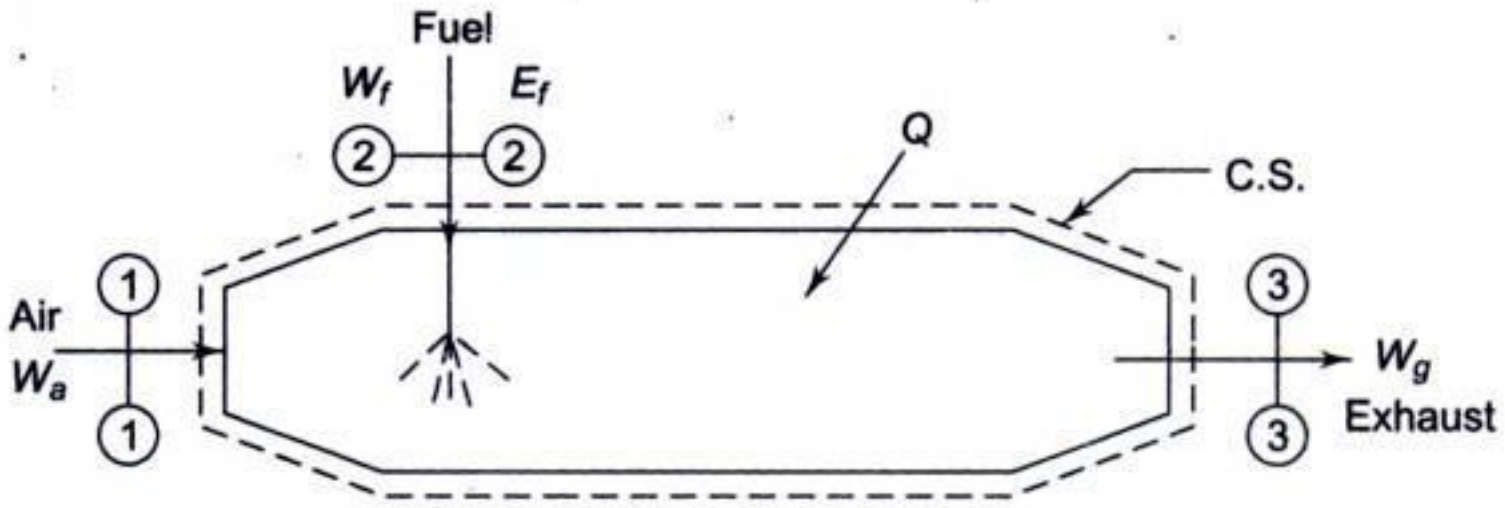

Fig. 5.19

Example 5.8 In a reciprocating engine, the mass of gas occupying the clearance volume is m_c kg at state p_1, u_1, v_1 and h_1. By opening the inlet valve, m_f kg of gas is taken into the cylinder, and at the conclusion of the intake process the state of the gas is given by p_2, u_2, v_2, h_2. The state of the gas in the supply pipe is constant and is given by $p_p, u_p, v_p, h_p, \mathbf{V}_p$. How much heat is transferred between the gas and the cylinder walls during the intake process?

Solution Let us consider the control volume as shown in Fig. 5.20. Writing the energy balance on a time rate basis

$$\frac{dE_V}{d\tau}=\frac{đQ}{d\tau}-\frac{đW}{d\tau}+\left(h_p+\frac{\mathbf{V}_p^2}{2}\right)\frac{dm}{d\tau}$$

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(ii) The charging process will stop when pressure inside the vessel reaches that of the main. The charging time can be found by setting $p = p_1$ in the pressure relation

$$p_1 - p_0 = \frac{\dot{m}_0 R\gamma T_1}{aV} - \frac{\dot{m}_0 R}{aV} e^{-a\tau} \gamma T_1$$

By rearrangement,

$$e^{a\tau} = \frac{\dot{m}_0 R\gamma T_1/(aV)}{\left(\frac{m_0 R\gamma T_1}{aV}\right) - (p_1 - p_0)}$$

$$\therefore \qquad \tau = \frac{1}{a} \ln \left[1 - (p_1 - p_0) \frac{aV}{m_0 R\gamma T_1}\right]$$

Review Questions

5.1 Explain the system approach and the control volume approach in the analysis of a flow process.
5.2 What is a steady flow process?
5.3 Write the steady flow energy equation for a single stream entering and a single stream leaving a control volume and explain the various terms in it.
5.4 Give the differential form of the S.F.E.E.
5.5 Under what conditions does the S.F.E.E. reduce to Euler's equation?
5.6 How does Bernoulli's equation compare with S.F.E.E.?
5.7 What will be the velocity of a fluid leaving a nozzle, if the velocity of approach is very small?
5.8 Show that the enthalpy of a fluid before throttling is equal to that after throttling.
5.9 Write the general energy equation for a variable flow process.
5.10 What is the system technique in a bottle-filling process?
5.11 Explain the control volume technique in a variable flow process.

Problems

5.1 A blower handles 1 kg/s of air at 20°C and consumes a power of 15 kW. The inlet and outlet velocities of air are 100 m/s and 150 m/s respectively. Find the exit air temperature, assuming adiabatic conditions. Take c_p of air is 1.005 kJ/kg-K. Ans. 28.38°C

5.2 A turbine operates under steady flow conditions, receiving steam at the following state: pressure 1.2 MPa, temperature 188°C, enthalpy 2785 kJ/kg, velocity 33.3 m/s and elevation 3 m. The steam leaves the turbine at the following state: pressure 20 kPa, enthalpy 2512 kJ/kg, velocity 100 m/s, and elevation 0 m. Heat is lost to the surroundings at the rate of 0.29 kJ/s. If the rate of steam flow through

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(a) An evacuated bottle is fitted with a valve through which air from the atmosphere, at 760 mm Hg and 25°C, is allowed to flow slowly to fill the bottle. If no heat is transferred to or from the air in the bottle, what will its temperature be when the pressure in the bottle reaches 760 mm Hg?

Ans. 144.2°C

(b) If the bottle initially contains 0.03 m^3 of air at 400 mm Hg and 25°C, what will the temperature be when the pressure in the bottle reaches 760 mm Hg?

Ans. 71.6°C

5.20 A pressure cylinder of volume V contains air at pressure p_0 and temperature T_0. It is to be filled from a compressed air line maintained at constant pressure p_1 and temperature T_1. Show that the temperature of the air in the cylinder after it has been charged to the pressure of the line is given by

$$T = \frac{\gamma T_1}{1 + \frac{p_0}{p_1}\left(\gamma \frac{T_1}{T_0} - 1\right)}$$

5.21 A small reciprocating vacuum pump having the rate of volume displacement V_d is used to evacuate a large vessel of volume V. The air in the vessel is maintained at a constant temperature T by energy transfer as heat. If the initial and final pressures are p_1 and p_2 respectively, find the time taken for the pressure drop and the necessary energy transfer as heat during evacuation. Assume that for air, $pV = mRT$, where m is the mass and R is a constant, and u is a function of T only.

$$\left[\text{Ans. } t = \frac{V}{V_d} \ln \frac{p_1}{p_2}; \quad Q = (p_1 - p_2)V\right]$$

[Hint: $dm = -p(V_d \cdot dt)/(RT) = V\,dp/(RT)$].

5.22 A tank containing 45 kg of water initially at 45°C has one inlet and one exit with equal mass flow rates. Liquid water enters at 45°C and a mass flow rate of 270 kg/h. A cooling coil immersed in the water removes energy at the ate of 7.6 kW. The water is well mixed by a paddle wheel with a power input of 0.6 kW. The pressures at inlet and exit are equal. Ignoring changes in KE and PE, find the variation of water temperature with time.

Ans. $T = 318 - 22\,[1 - \exp(-6t)]$

5.23 A rigid tank of volume 0.5 m^3 is initially evacuated. A tiny hole develops in the wall, and air from the surroundings at 1 bar, 21°C leaks in. Eventually, the pressure in the tank reaches 1 bar. The process occurs slowly enough that heat transfer between the tank and the surroundings keeps the temperature of the air inside the tank constant at 21°C. Determine the amount of heat transfer.

Ans. – 50 kJ

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a certain quantity of water undergoing the energy interactions, as shown, in cyclic operations to produce net work from a certain heat input.

The function of a heat engine cycle is to produce work continuously at the expense of heat input to the system. So the net work W_{net} and heat input Q_1 referred to the cycle are of primary interest. The efficiency of a heat engine or a heat engine cycle is defined as follows:

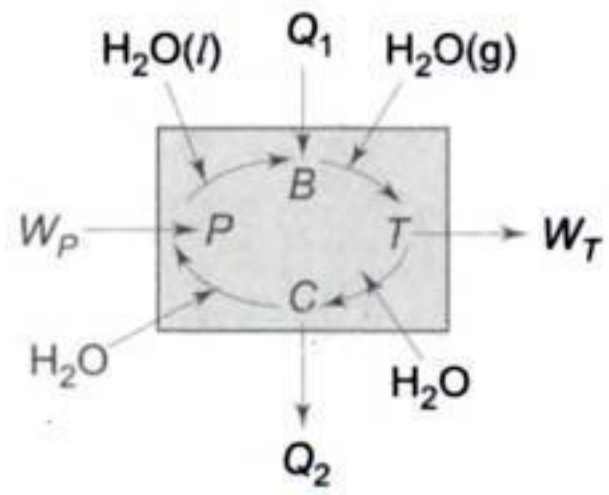

Fig. 6.3 *Cyclic Heat Engine with Energy Interactions Represented in a Block Diagram*

$$\eta = \frac{\text{Net work output of the cycle}}{\text{Total heat input to the cycle}}$$

$$= \frac{W_{net}}{Q_1} \qquad (6.4)$$

From Eqs (6.1), (6.2), (6.3) and (6.4),

$$\eta = \frac{W_{\text{net}}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$

$$\eta = 1 - \frac{Q_2}{Q_1} \qquad (6.5)$$

This is also known as the *thermal efficiency* of a heat engine cycle. A heat engine is very often called upon to extract as much work (net) as possible from a certain heat input, i.e. to maximize the cycle efficiency.

6.3 ENERGY RESERVOIRS

A thermal energy reservoir (TER) is defined as a large body of infinite heat capacity, which is capable of absorbing or rejecting an unlimited quantity of heat without suffering appreciable changes in its thermodynamic coordinates. The changes that do take place in the large body as heat enters or leaves are so very slow and so very minute that all processes within it are quasi-static.

The thermal energy reservoir TER_H from which heat Q_1 is transferred to the system operating in a heat engine cycle is called the *source*. The thermal energy reservoir TER_L to which heat Q_2 is rejected from the system during a cycle is the *sink*. A typical source is a constant temperature furnace where fuel is continuously burnt, and a typical sink is a river or sea or the atmosphere itself.

A mechanical energy reservoir (MER) is a large body enclosed by an adiabatic impermeable wall capable of storing work as potential energy (such as a raised weight or wound spring) or kinetic energy (such as a rotating flywheel). All processes of interest within an MER are essentially quasi-static. An MER receives and delivers mechanical energy quasi-statically.

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$$\therefore \quad [\text{COP}]_{H.P.} = \frac{Q_1}{Q_1 - Q_2} \qquad (6.7)$$

From Eqs. (6.6) and (6.7), it is found that

$$[\text{COP}]_{H.P.} = [\text{COP}]_{\text{ref}} + 1 \qquad (6.8)$$

The COP of a heat pump is greater than the COP of a refrigerator by unity. Equation (6.8) expresses a very interesting feature of a heat pump. Since

$$Q_1 = [\text{COP}]_{H.P.}\, W$$
$$= [\text{COP}_{\text{ref}} + 1]\, W \qquad (6.9)$$

Q_1 is always greater than W.

For an electrical resistance heater, if W is the electrical energy consumption, then the heat transferred to the space at steady state is W only, i.e. $Q_1 = W$.

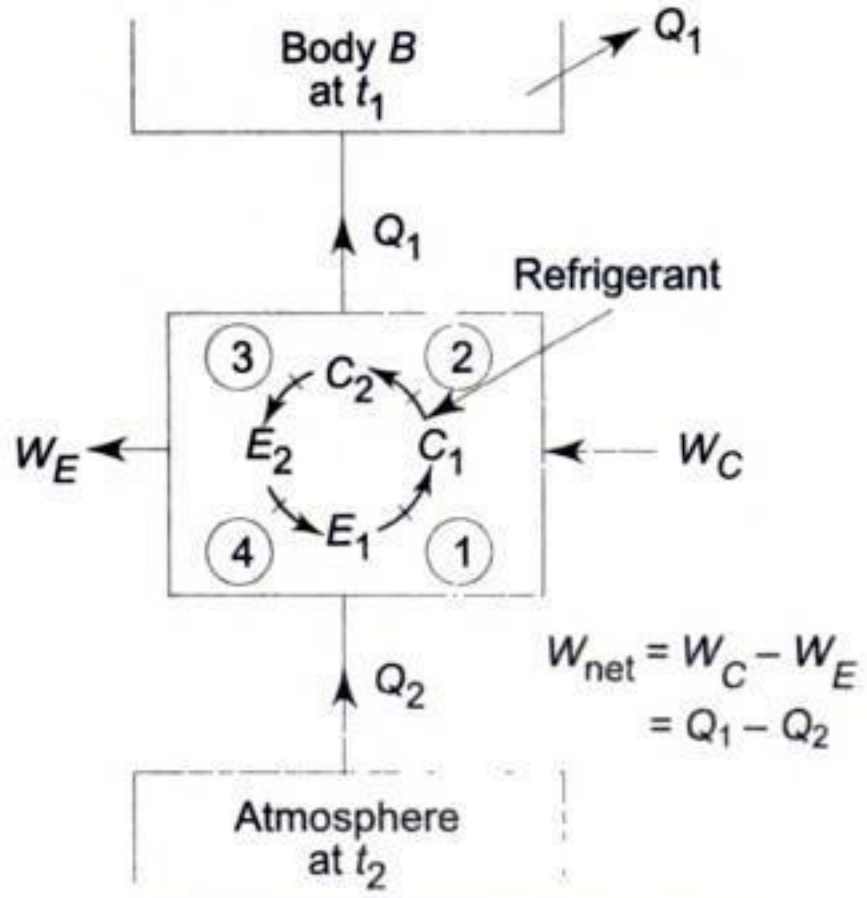

Fig. 6.8 *A Cyclic Heat Pump*

A 1 kW electric heater can give 1 kW of heat at steady state and nothing more. In other words, 1 kW of work (high grade energy) dissipates to give 1 kW of heat (low grade energy), which is thermodynamically inefficient.

However, if this electrical energy W is used to drive the compressor of a heat pump, the heat supplied Q_1 will always be more than W, or $Q_1 > W$. Thus, a heat pump provides a thermodynamic advantage over direct heating.

For heat to flow from a cooler to a hotter body, W cannot be zero, and hence, the COP (both for refrigerator and heat pump) cannot be infinity. Therefore, $W > 0$, and COP $< \infty$.

6.7 EQUIVALENCE OF KELVIN-PLANCK AND CLAUSIUS STATEMENTS

At first sight, Kelvin-Planck's and Clausius' statements may appear to be unconnected, but it can easily be shown that they are virtually two parallel statements of the second law and are equivalent in all respects.

The equivalence of the two statements will be proved if it can be shown that the violation of one statement implies the violation of the second, and vice versa.

(a) Let us first consider a cyclic heat pump P which transfers heat from a low temperature reservoir (t_2) to a high temperature reservoir (t_1) with no other effect, i.e. with no expenditure of work, violating Clausius statement (Fig. 6.9).

Let us assume a cyclic heat engine E operating between the same thermal energy reservoirs, producing W_{net} in one cycle. The rate of working of the heat engine is such that it draws an amount of heat Q_1 from the hot reservoir equal to that discharged by the heat pump. Then the hot reservoir may be eliminated and the heat Q_1 discharged by the heat pump is fed to the heat engine. So we see that the heat pump P and the heat engine E acting together constitute a heat engine operating in cycles and producing net work while exchanging heat only with one body at a single fixed temperature. This violates the Kelvin-Planck statement.

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(b) Lack of Pressure Equilibrium within the Interior of the System or between the System and the Surroundings

When there exists a difference in pressure between the system and the surroundings, or within the system itself, then both the system and its surroundings or the system alone, will undergo a change of state which will cease only when mechanical equilibrium is established. The reverse of this process is not possible spontaneously without producing any other effect. That the reverse process will violate the second law becomes obvious from the following illustration.

(c) Free Expansion

Let us consider an insulated container (Fig. 6.14) which is divided into two compartments *A* and *B* by a thin diaphragm. Compartment *A* contains a mass of gas, while compartment *B* is completely evacuated. If the diaphragm is punctured, the gas in *A* will expand into *B* until the pressures in *A* and *B* become equal. This is known as free or unrestrained expansion. We can demonstrate by the second law, that the process of free expansion is irreversible.

To prove this, let us assume that free expansion is reversible, and that the gas in *B* returns into *A* with an increase in pressure, and *B* becomes evacuated as before (Fig. 6.15). There is no other effect. Let us install an engine (a machine, not a cyclic heat engine) between *A* and *B*, and permit the gas to expand through the engine from *A* to *B*. The engine develops a work output *W* at the expense of the internal energy of the gas. The internal energy of the gas (system) in *B* can be restored to its initial value by heat transfer Q (= W) from a source. Now, by the use of the reversed free expansion, the system can be restored to the initial state of high pressure in *A* and vacuum in *B*. The net result is a cycle, in which we observe that net work output *W* is accomplished by exchanging heat with a single reservoir. This violates the Kelvin-Planck statement. Hence, free expansion is irreversible.

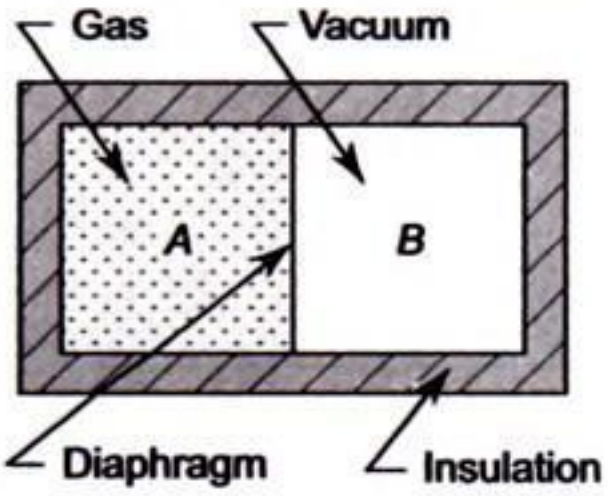

Fig. 6.14 *Free Expansion*

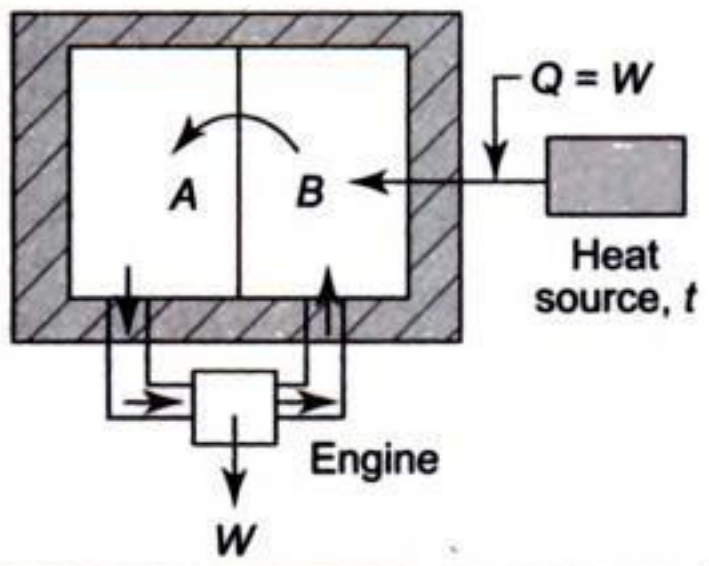

Fig. 6.15 *Second Law Demonstrates that Free Expansion is Irreversible*

The same argument will hold if the compartment *B* is not in vacuum but at a pressure lower than that in compartment *A* (case b).

6.9.2 Irreversibility due to Dissipative Effects

The irreversibility of a process may be due to the *dissipative effects* in which work is done without producing an equivalent increase in the kinetic or potential

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system, as in a piston and cylinder machine, the cycle consists of the following four successive processes (Fig. 6.20):

(a) *A reversible isothermal process* in which heat Q_1 enters the system at t_1 reversibly from a constant temperature source at t_1 when the cylinder cover is in contact with the diathermic cover *A*. The internal energy of the system increases.

From First law,

$$Q_1 = U_2 - U_1 + W_{1\text{-}2} \tag{6.14}$$

(for an ideal gas only, $U_1 = U_2$)

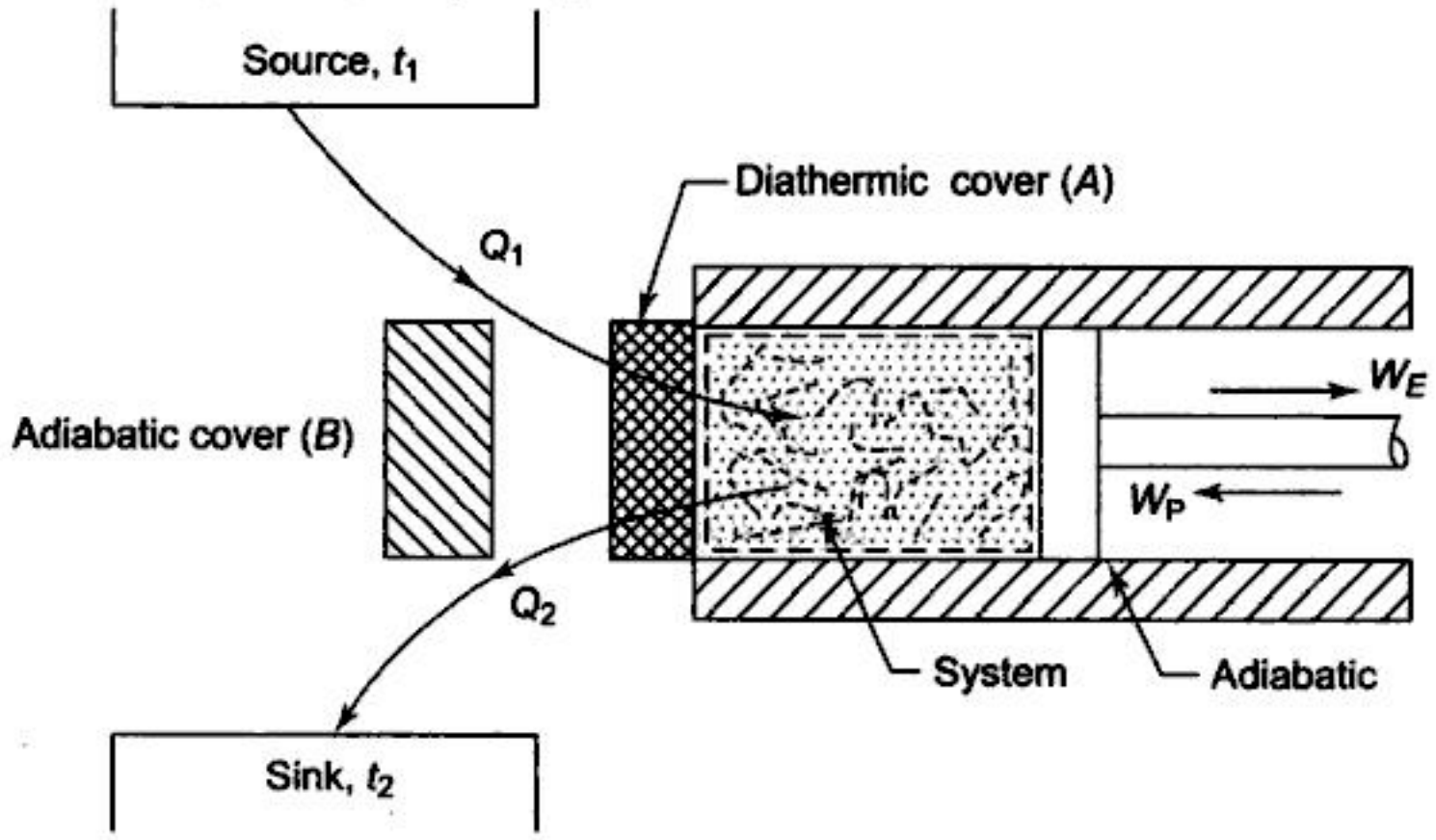

Fig. 6.20 *Carnot Heat Engine—Stationary System*

(b) *A reversible adiabatic process* in which the diathermic cover *A* is replaced by the adiabatic cover *B*, and work W_E is done by the system adiabatically and reversibly at the expense of its internal energy, and the temperature of the system decreases from t_1 to t_2.

Using the first law,

$$0 = U_3 - U_2 + W_{2-3} \tag{6.15}$$

(c) *A reversible isothermal process* in which *B* is replaced by *A* and heat Q_2 leaves the system at t_2 to a constant temperature sink at t_2 reversibly, and the internal energy of the system further decreases.

From the first law,

$$-Q_2 = U_4 - U_3 - W_{3-4} \tag{6.16}$$

only for an ideal gas, $U_3 = U_4$

(d) *A reversible adiabatic process* in which *B* again replaces *A*, and work W_p is done upon the system reversibly and adiabatically, and the internal energy of the system increases and the temperature rises from t_2 to t_1.

Applying the first law,

$$0 = U_1 - U_4 - W_{4-1} \tag{6.17}$$

Two reversible isotherms and two reversible adiabatics constitute a Carnot cycle, which is represented in *p-v* coordinates in Fig. 6.21.

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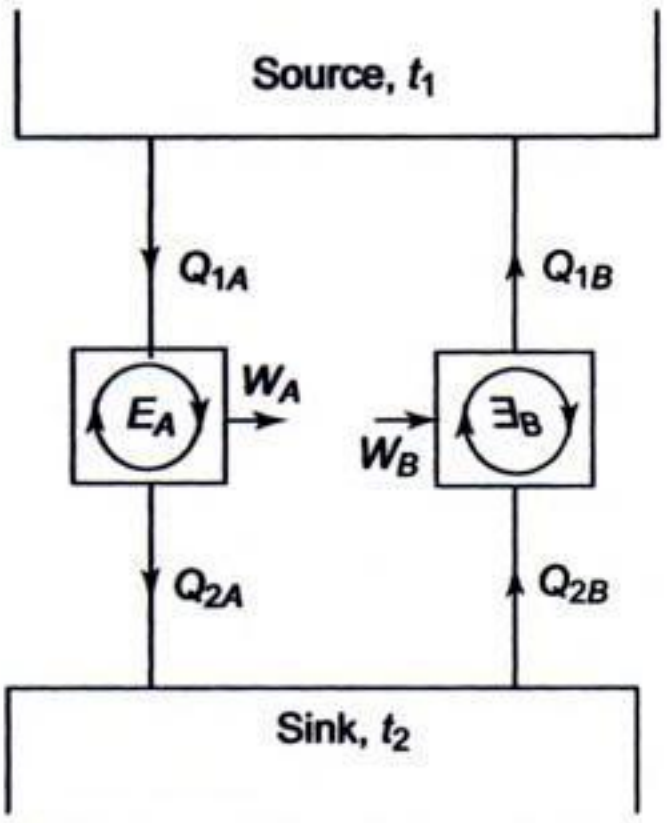

Fig. 6.26 E_B is Reversed

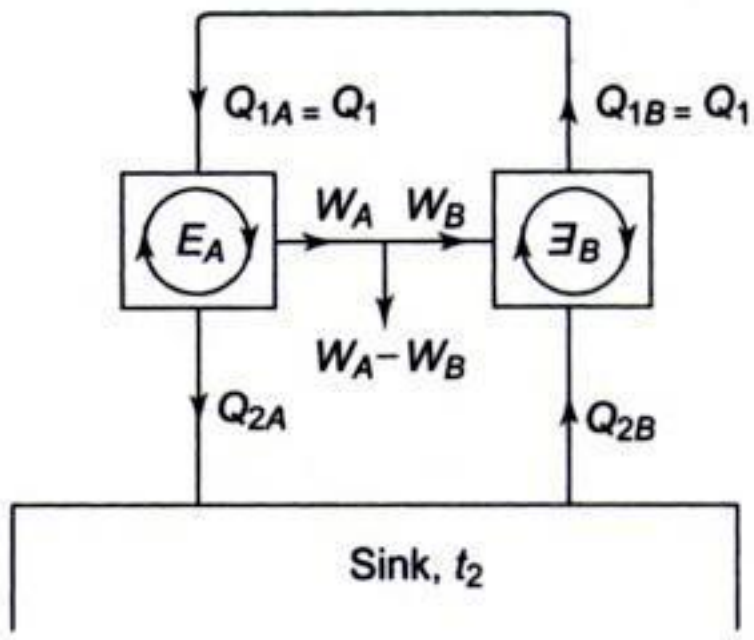

Fig. 6.27 E_A and $\exists_B$ together Violate the K-P Statement

6.14 COROLLARY OF CARNOT'S THEOREM

The efficiency of all reversible heat engines operating between the same temperature levels is the same.

Let both the heat engines E_A and E_B (Fig. 6.25) be reversible. Let us assume $\eta_A > \eta_B$. Similar to the procedure outlined in the preceding article, if E_B is reversed to run, say, as a heat pump using some part of the work output (W_A) of engine E_A, we see that the combined system of heat pump E_B and engine E_A, becomes a PMM2. So η_A cannot be greater than η_B. Similarly, if we assume $\eta_B > \eta_A$ and reverse the engine E_A, we observe that η_B cannot be greater than η_A.

Therefore $$\eta_A = \eta_B$$

Since the efficiencies of all reversible heat engines operating between the same heat reservoirs are the same, *the efficiency of a reversible engine is independent of the nature or amount of the working substance undergoing the cycle.*

6.15 ABSOLUTE THERMODYNAMIC TEMPERATURE SCALE

The efficiency of any heat engine cycle receiving heat Q_1 and rejecting heat Q_2 is given by

$$\eta = \frac{W_{net}}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} \tag{6.18}$$

By the second law, it is necessary to have a temperature difference $(t_1 - t_2)$ to obtain work for any cycle. We know that the efficiency of all heat engines operating between the same temperature levels is the same, and it is independent of the working substance. Therefore, for a reversible cycle (Carnot cycle), the efficiency will depend solely upon the temperatures t_1 and t_2, at which heat is transferred, or

$$\eta_{rev} = f(t_1, t_2) \tag{6.19}$$

where f signifies some function of the temperatures. From Equations (6.18) and (6.19)

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Thus any attainable value of absolute temperature is always greater than zero. This is also known as the *Third Law of Thermodynamics* which may be stated as follows: *It is impossible by any procedure, no matter how idealized, to reduce any system to the absolute zero of temperature in a finite number of operations.*

This is what is called the Fowler-Guggenheim statement of the third law. The third law itself is an independent law of nature, and not an extension of the second law. The concept of heat engine is not necessary to prove the non-attainability of absolute zero of temperature by any system in a finite number of operations.

6.16 EFFICIENCY OF THE REVERSIBLE HEAT ENGINE

The efficiency of a reversible heat engine in which heat is received solely at T_1 is found to be

$$\eta_{rev} = \eta_{max} = 1 - \left(\frac{Q_2}{Q_1}\right)_{rev} = 1 - \frac{T_2}{T_1}$$

or

$$\eta_{rev} = \frac{T_1 - T_2}{T_1}$$

It is observed here that as T_2 decreases, and T_1 increases, the efficiency of the reversible cycle increases.

Since η is always less than unity, T_2 is always greater than zero and positive.

The COP of a refrigerator is given by

$$(\text{COP})_{refr} = \frac{Q_2}{Q_1 - Q_2} = \frac{1}{\frac{Q_1}{Q_2} - 1}$$

For a reversible refrigerator, using

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

$$[\text{COP}_{refr}]_{rev} = \frac{T_2}{T_1 - T_2} \tag{6.24}$$

Similarly, for a reversible heat pump

$$[\text{COP}_{H.P.}]_{rev} = \frac{T_1}{T_1 - T_2} \tag{6.25}$$

6.17 EQUALITY OF IDEAL GAS TEMPERATURE AND KELVIN TEMPERATURE

Let us consider a Carnot cycle executed by an ideal gas, as shown in Fig. 6.31.

The two isothermal processes *a–b* and *c–d* are represented by equilateral hyperbolas whose equations are respectively

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Solution Freezer temperature,

$$T_2 = -15 + 273 = 258 \text{ K}$$

Ambient air temperature,

$$T_1 = 30 + 273 = 303 \text{ K}$$

The refrigerator cycle removes heat from the freezer at the same rate at which heat leaks into it (Fig. 6.33).

For minimum power requirement

$$\frac{Q_2}{T_2} = \frac{Q_1}{T_1}$$

$$\therefore \quad Q_1 = \frac{1.75}{2.8} \times 303 = 2.06 \text{ kJ/s}$$

$$\therefore \quad W = Q_1 - Q_2$$

$$= 2.06 - 1.75 = 0.31 \text{ kJ/s} = \mathbf{0.31\ kW}$$

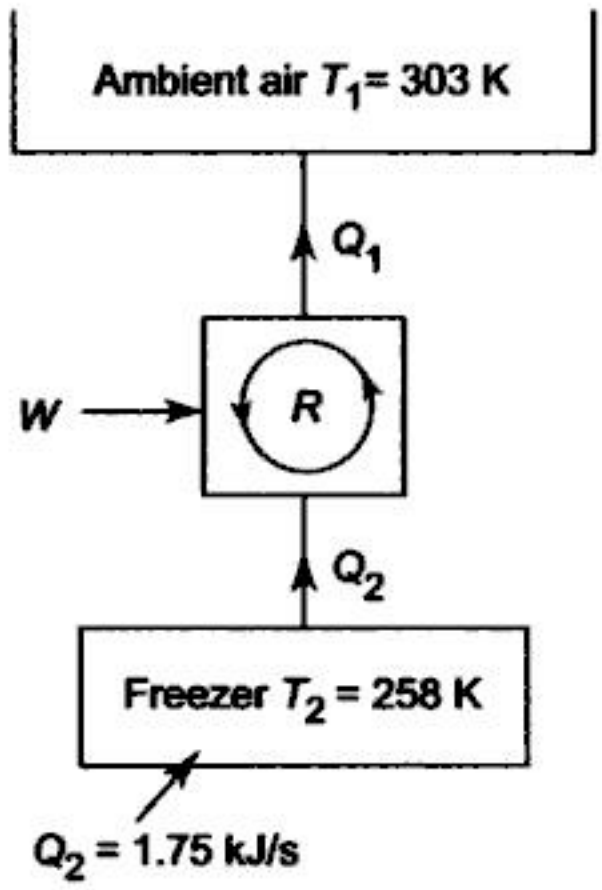

Fig. 6.33

Example 6.3 A reversible heat engine operates between two reservoirs at temperatures of 600°C and 40°C. The engine drives a reversible refrigerator which operates between reservoirs at temperatures of 40°C and – 20°C. The heat transfer to the heat engine is 2000 kJ and the net work output of the combined engine refrigerator plant is 360 kJ.

(a) Evaluate the heat transfer to the refrigerant and the net heat transfer to the reservoir at 40°C.

(b) Reconsider (a) given that the efficiency of the heat engine and the COP of the refrigerator are each 40% of their maximum possible values.

Solution (a) Maximum efficiency of the heat engine cycle (Fig. 6.34) is given by

$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{313}{873} = 1 - 0.358 = 0.642$$

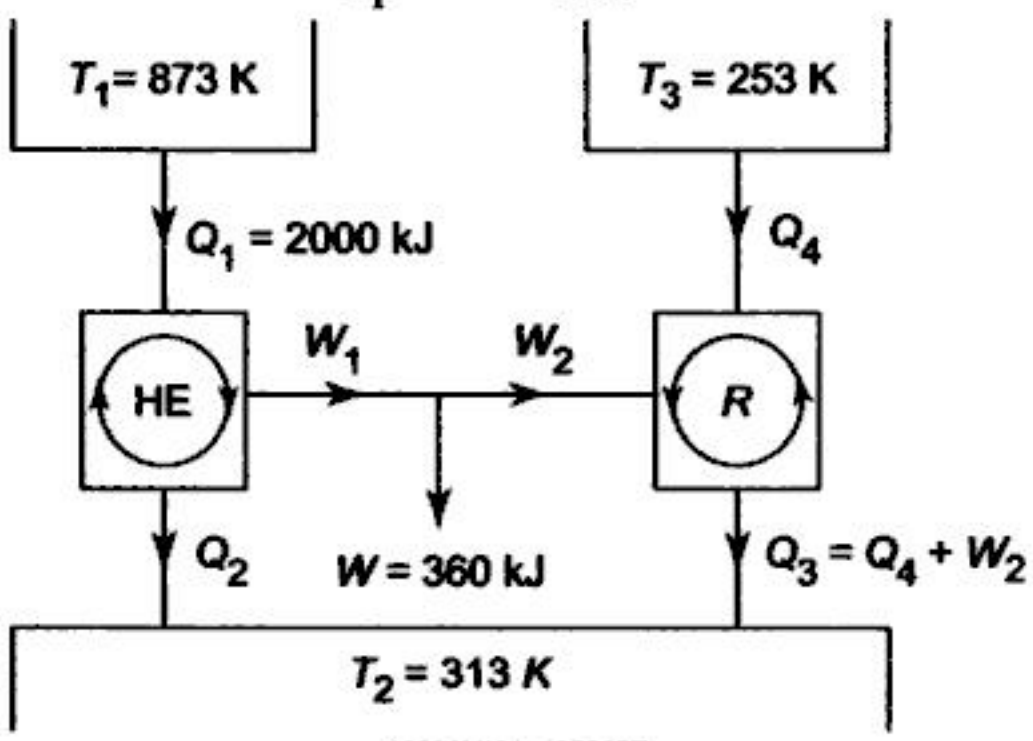

Fig. 6.34

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or $$\text{M.F.} = \frac{T_2^2 - T_2T_3 + T_2T_1 - T_2^2}{T_1(T_2 - T_3)}$$

or $$\text{M.F.} = \frac{T_2(T_1 - T_3)}{T_1(T_2 - T_3)}$$

Here $T_1 = 473$ K, $T_2 = 293$ K and $T_3 = 273$ K

$\therefore$ $$\text{M.F.} = \frac{293(473-273)}{473(293-273)} = \frac{2930}{473} = \mathbf{6.3}$$

which means that every kg of coal burned would deliver the heat equivalent to over 6 kg. Of course, in an actual case, the efficiencies would be less than Carnot efficiencies, but even with a reduction of 50%, the possible savings would be quite significant.

Example 6.6 It is proposed that solar energy be used to warm a large collector plate. This energy would, in turn, be transferred as heat to a fluid within a heat engine, and the engine would reject energy as heat to the atmosphere. Experiments indicate that about 1880 kJ/m^2 h of energy can be collected when the plate is operating at 90°C. Estimate the minimum collector area that would be required for a plant producing 1 kW of useful shaft power. The atmospheric temperature may be assumed to be 20°C.

Solution The maximum efficiency for the heat engine operating between the collector plate temperature and the atmospheric temperature (Fig. 6.38) as follows:

$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{293}{363} = 0.192$$

The efficiency of any actual heat engine operating between these temperatures would be less than this efficiency.

$\therefore$ $$Q_{min} = \frac{W}{\eta_{max}} = \frac{1 \text{ kJ/s}}{0.192} = 5.21 \text{ kJ/s}$$

$$= 18{,}800 \text{ kJ/h}$$

$\therefore$ Minimum area required for the collector plate

$$= \frac{18{,}800}{1880} = \mathbf{10 \text{ m}^2}$$

Example 6.7 A reversible heat engine in a satellite operates between a hot reservoir at T_1 and a radiating panel at T_2. Radiation from the panel is proportional to its area and to T_2^4. For a given work output and value of T_1 show that the area of the panel will be minimum when $\frac{T_2}{T_1} = 0.75$.

Determine the minimum area of the panel for an output of 1 kW if the constant of proportionality is 5.67×10^{-8} W/m^2 K^4 and T_1 is 1000 K.

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6.5 A household refrigerator is maintained at a temperature of 2°C. Every time the door is opened, warm material is placed inside, introducing an average of 420 kJ, but making only a small change in the temperature of the refrigerator. The door is opened 20 times a day, and the refrigerator operates at 15% of the ideal COP. The cost of work is Rs. 2.50 per kWh. What is the monthly bill for this refrigerator? The atmosphere is at 30°C. Ans Rs. 118.80

6.6 A heat pump working on the Carnot cycle takes in heat from a reservoir at 5°C and delivers heat to a reservoir at 60°C. The heat pump is driven by a reversible heat engine which takes in heat from a reservoir at 840°C and rejects heat to a reservoir at 60°C. The reversible heat engine also drives a machine that absorbs 30 kW. If the heat pump extracts 17 kJ/s from the 5°C reservoir, determine (a) the rate of heat supply from the 840°C source, and (b) the rate of heat rejection to the 60°C sink. Ans. (a) 47.61 kW; (b) 34.61 kW

6.7 A refrigeration plant for a food store operates as a reversed Carnot heat engine cycle. The store is to be maintained at a temperature of – 5°C and the heat transfer from the store to the cycle is at the rate of 5 kW. If heat is transferred from the cycle to the atmosphere at a temperature of 25°C, calculate the power required to drive the plant. Ans. 0.56 kW

6.8 A heat engine is used to drive a heat pump. The heat transfers from the heat engine and from the heat pump are used to heat the water circulating through the radiators of a building. The efficiency of the heat engine is 27% and the COP of the heat pump is 4. Evaluate the ratio of the heat transfer to the circulating water to the heat transfer to the heat engine. Ans. 1.81

6.9 If 20 kJ are added to a Carnot cycle at a temperature of 100°C and 14.6 kJ are rejected at 0°C, determine the location of absolute zero on the Celsius scale. Ans. – 270.37°C

6.10 Two reversible heat engines *A* and *B* are arranged in series, A rejecting heat directly to *B*. Engine *A* receives 200 kJ at a temperature of 421°C from a hot source, while engine *B* is in communication with a cold sink at a temperature of 4.4°C. If the work output of *A* is twice that of *B*, find (a) the intermediate temperature between *A* and *B*, (b) the efficiency of each engine, and (c) the heat rejected to the cold sink. Ans. 143.4°C, 40% and 33.5%, 80 kJ

6.11 A heat engine operates between the maximum and minimum temperatures of 671°C and 60°C respectively, with an efficiency of 50% of the appropriate Carnot efficiency. It drives a heat pump which uses river water at 4.4°C to heat a block of flats in which the temperature is to be maintained at 21.1°C. Assuming that a temperature difference of 11.1°C exists between the working fluid and the river water, on the one hand, and the required room temperature on the other, and assuming the heat pump to operate on the reversed Carnot cycle, but with a COP of 50% of the ideal COP, find the heat input to the engine per unit heat output from the heat pump. Why is direct heating thermodynamically more wasteful? Ans. 0.79 kJ/kJ heat input

6.12 An ice-making plant produces ice at atmospheric pressure and at 0°C from water. The mean temperature of the cooling water circulating through the condenser of the refrigerating machine is 18°C. Evaluate the minimum electrical work in kWh required to produce 1 tonne of ice. (The enthalpy of fusion of ice at atmospheric pressure is 333.5 kJ/kg). Ans. 6.11 kWh

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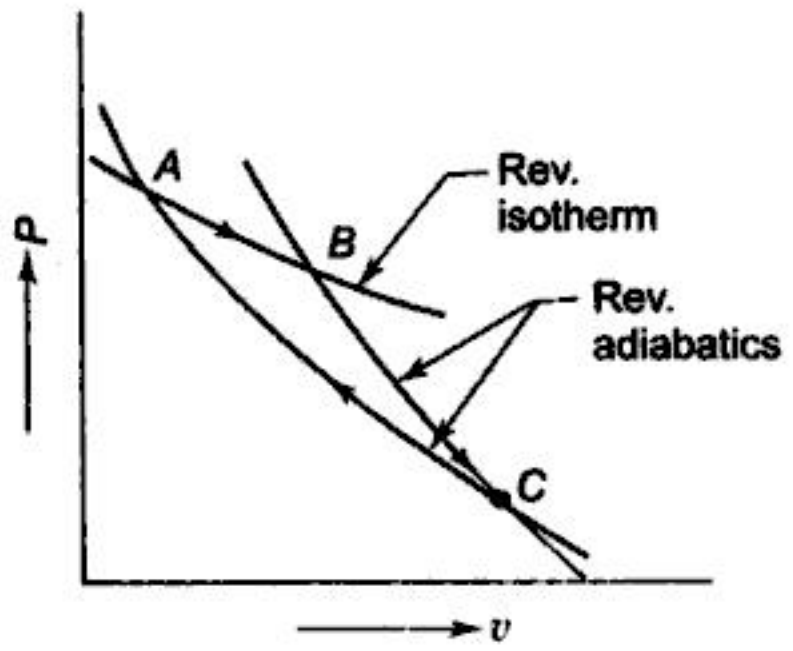

Fig. 7.1 *Assumption of Two Reversible Adiabatics Intersecting Each Other*

7.3 CLAUSIUS' THEOREM

Let a system be taken from an equilibrium state i to another equilibrium state f by following the reversible path i-f (Fig. 7.2). Let a reversible adiabatic i-a be drawn through i and another reversible adiabatic b-f be drawn through f. Then a reversible isotherm a-b is drawn in such a way that the area under i-a-b-f is equal to the area under i-f. Applying the first law for

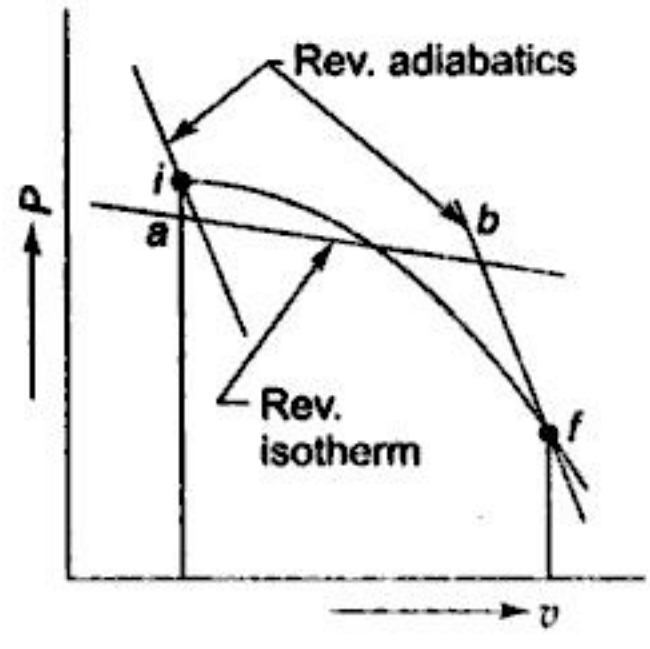

Fig. 7.2 *Reversible Path Substituted by Two Reversible Adiabatics and a Reversible Isotherm*

Process i-f

$$Q_{i\text{-}f} = U_f - U_i + W_{if} \qquad (7.1)$$

Process i-a-b-f

$$Q_{iabf} = U_f - U_i + W_{iabf} \qquad (7.2)$$

Since

$$W_{if} = W_{iabf}$$

$\therefore$ From Eqs. (7.1) and (7.2)

$$Q_{if} = Q_{iabf}$$
$$= Q_{ia} + Q_{ab} + Q_{bf}$$

Since $Q_{ia} = 0$ and $Q_{bf} = 0$

$$Q_{if} = Q_{ab}$$

Heat transferred in the process i-f is equal to the heat transferred in the isothermal process a-b.

Thus any reversible path may be substituted by a reversible zigzag path, between the same end states, consisting of a reversible adiabatic followed by a reversible isotherm and then by a reversible adiabatic, such that the heat transferred during the isothermal process is the same as that transferred during the original process.

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For reversible isothermal heat transfer (Fig. 7.7), T = constant.

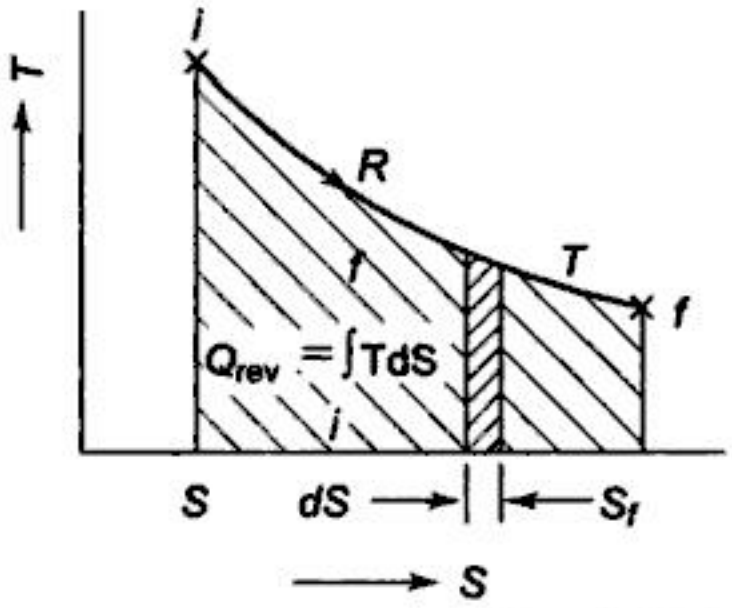

Fig. 7.6 *Area Under a Reversible Path on the T-s Plot Represents Heat Transfer*

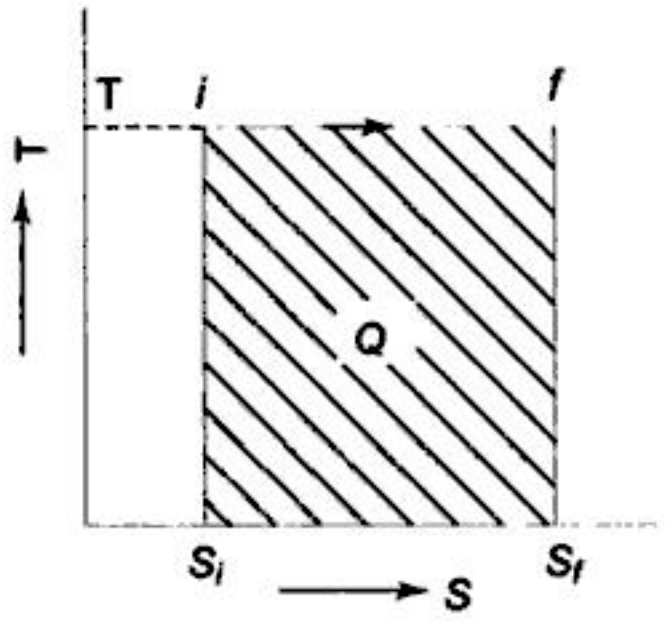

Fig. 7.7 *Reversible Isothermal Heat Transfer*

$$\therefore \qquad Q_{rev} = T\int_i^f dS = T\,(S_f - S_i)$$

For a reversible adiabatic process, $dS = 0$, $S = C$ (Fig. 7.8).

The *Carnot cycle* comprising two reversible isotherms and two reversible adiabatics forms a rectangle in the *T-S* plane (Fig. 7.9). Process 4 – 1 represents reversible isothermal heat addition Q_1 to the system at T_1 from an external source, process 1–2 is the reversible adiabatic expansion of the system producing W_E amount of work, process 2 – 3 is the reversible isothermal heat rejection from the system to an external sink at T_2, and process 3 – 4 represents reversible adiabatic compression of the system consuming W_c amount of work. Area 1234 represents the net work output per cycle and the area under 4 – 1 indicates the quantity of heat added to the system Q_1.

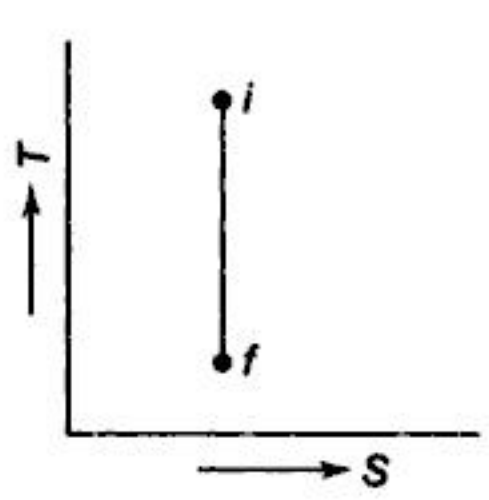

Fig. 7.8 *Reversible Adiabatic is Isentropic*

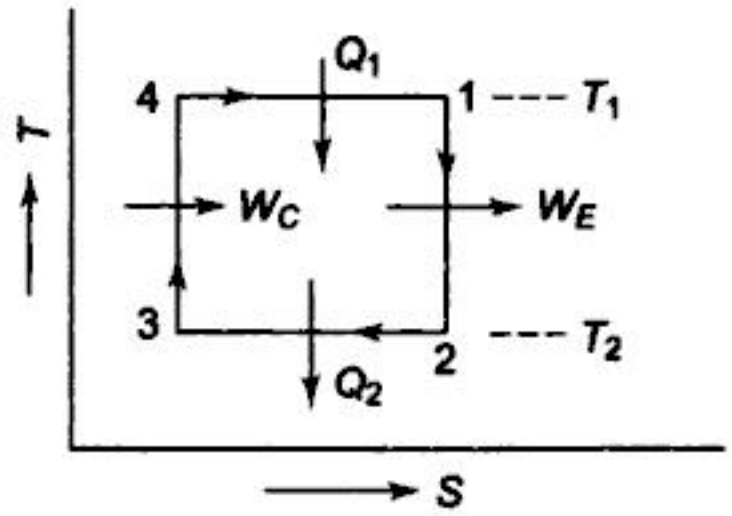

Fig. 7.9 *Carnot Cycle*

$$\therefore \qquad \eta_{Carnot} = \frac{Q_1 - Q_2}{Q_1} = \frac{T_1\,(S_1 - S_4) - T_2\,(S_2 - S_3)}{T_1\,(S_1 - S_4)}$$

$$= \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}$$

and $$W_{net} = Q_1 - Q_2 = (T_1 - T_2)\,(S_1 - S_4)$$

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Entropy change for the fluid in subsystem 1

$$\Delta S_1 = \int_{T_1}^{T_f} \frac{dQ_{rev}}{T} = \int_{T_f}^{T_f} \frac{m_1 c_1 dT}{T} = m_1 c_1 \ln \frac{T_f}{T_1}$$

$$= m_1 c_1 \ln \frac{t_f + 273}{t_1 + 273}$$

This will be negative, since $T_1 > T_f$

Entropy change for the fluid in subsystem 2

$$\Delta S_2 = \int_{T_2}^{T_f} \frac{m_2 c_2 dT}{T} = m_2 c_2 \ln \frac{T_f}{T_2} = m_2 c_2 \ln \frac{t_f + 273}{t_2 + 273}$$

This will be positive, since $T_2 < T_f$

$$\therefore \quad \Delta S_{univ} = \Delta S_1 + \Delta S_2$$

$$= m_1 c_1 \ln \frac{T_f}{T_1} + m_2 c_2 \ln \frac{T_f}{T_2}$$

ΔS_{univ} will be positive definite, and the mixing process is irreversible.

Although the mixing process is irreversible, to evaluate the entropy change for the subsystems, the irreversible path was replaced by a reversible path on which the integration was performed.

If $m_1 = m_2 = m$ and $c_1 = c_2 = c$.

$$\Delta S_{univ} = mc \ln \frac{T_f^2}{T_1 \cdot T_2}$$

and

$$T_f = \frac{m_1 c_1 T_1 + m_2 c_2 T_2}{m_1 c_1 + m_2 c_2} = \frac{T_1 + T_2}{2}$$

$$\therefore \quad \Delta S_{univ} = 2mc \ln \frac{(T_1 + T_2)/2}{\sqrt{T_1 \cdot T_2}}$$

This is always positive, since the arithmetic mean of any two numbers is always greater than their geometric mean. This can also be proved geometrically. Let a semi-circle be drawn with $(T_1 + T_2)$ as diameter (Fig. 7.16).

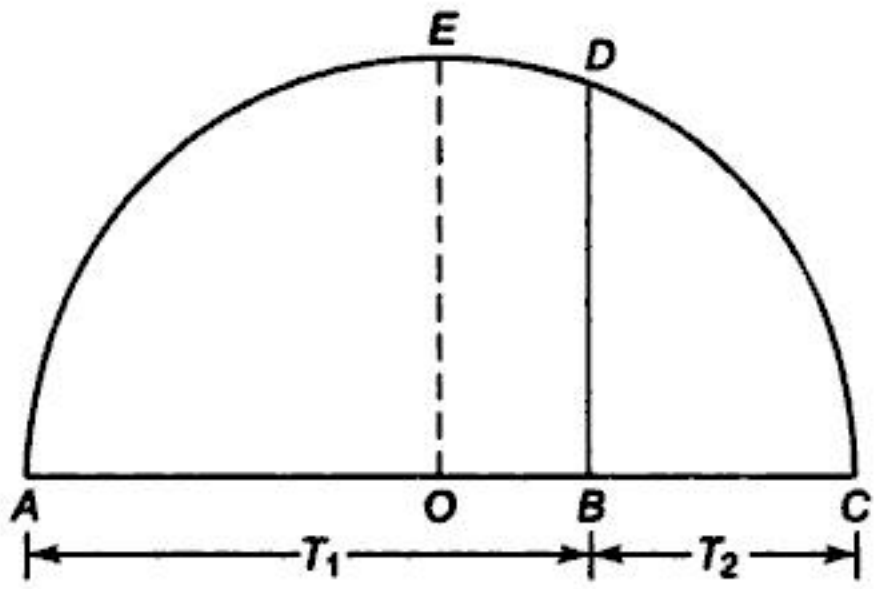

Fig. 7.16 *Geometrical Proof to Show That g.m. < a.m.*

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The *second law inequality* or the entropy principle gives

$$\underbrace{\sum_i \dot{m}_i S_i - \sum_e \dot{m}_e S_e}_{\text{net rate of entropy transfer}} + \frac{\dot{Q}}{T} \leq \underbrace{\frac{\partial S}{\partial \tau}}_{\text{rate of increase of entropy of the CV}} \tag{7.34}$$

Here $\dot{Q}$ represents the rate of heat transfer at the location of the boundary where the instantaneous temperature is T. The ratio $\dot{Q}/T$ accounts for the entropy transfer along with heat. The terms $\dot{m}_i S_i$ and $\dot{m}_e S_e$ account, respectively, for rates of entropy transfer into and out of the CV accompanying mass flow. The rate of entropy increase of the control volume exceeds, or is equal to, the net rate of entropy transfer into it. The difference is the entropy generated within the control volume due to irreversibility. Hence, the rate of entropy generation is given by

$$\dot{S}_{\text{gen}} = \frac{\partial S}{\partial \tau} - \sum_i \dot{m}_i S_i + \sum_e \dot{m}_e S_e - \frac{\dot{Q}}{T} \tag{7.35}$$

By the second law,

$$\dot{S}_{\text{gen}} \geq 0$$

If the process is reversible, $\dot{S}_{\text{gen}} = 0$. For an irreversible process, $\dot{S}_{\text{gen}} > 0$. The magnitude of $\dot{S}_{\text{gen}}$ quantifies the irreversibility of the process. If systems A and B operate so that $(\dot{S}_{\text{gen}})_A > (\dot{S}_{\text{gen}})_B$ it can be said that the system A operates more irreversibly than system B. The unit of $\dot{S}_{\text{gen}}$ is W/K.

As *steady state*, the continuity equation gives

$$\sum_i \dot{m}_i = \sum_e \dot{m}_e \tag{7.36}$$

the energy equation becomes

$$0 = \dot{Q} - W_{\text{sh}} + \sum_i \dot{m}_i \left(h + \frac{\mathbf{V}^2}{2} + gZ \right)_i - \sum_e \dot{m}_e \left(h + \frac{\mathbf{V}^2}{2} + gZ \right)_e \tag{7.37}$$

and the entropy equation reduces to

$$0 = \frac{\dot{Q}}{T} + \sum_i \dot{m}_i S_i - \sum_e \dot{m}_e s_e + \dot{S}_{\text{gen}} \tag{7.38}$$

These equations often must be solved simultaneously, together with appropriate property relations.

Mass and energy are conserved quantities, but entropy is not generally conserved. The rate at which entropy is transferred out must exceed the rate at which entropy enters the CV, the difference being the rate of entropy generated within the CV owing to irreversibilities.

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The reversible work done by a steady flow system (Fig. 7.27b) would be

$$W_{1-2} = -\int_1^2 v dp = \text{Area } 12\,ab\,1$$

7.15 ENTROPY AND DIRECTION: THE SECOND LAW—A DIRECTIONAL LAW OF NATURE

Since the entropy of an isolated system can never decrease, it follows that only those processes are possible in nature which would give an entropy increase for the system and the surroundings together (the universe). All spontaneous processes in nature occur only in one direction from a higher to a lower potential, and these are accompanied by an entropy increase of the universe. When the potential gradient is infinitesimal (or zero in the limit), the entropy change of the universe is zero, and the process is reversible. The second law indicates the direction in which a process takes place. *A process always occurs in such a direction as to cause an increase in the entropy of the universe*. The macroscopic change ceases only when the potential gradient disappears and the equilibrium is reached when the entropy of the universe assumes a maximum value. To determine the equilibrium state of an isolated system it is necessary to express the entropy as a function of certain properties of the system and then render the function a maximum. At equilibrium, the system (isolated) exists at the peak of the entropy-hill, and $dS = 0$ (Fig. 7.13).

7.16 ENTROPY AND DISORDER

Work is a macroscopic concept. Work involves order or the orderly motion of molecules, as in the expansion or compression of a gas. The kinetic energy and potential energy of a system represent orderly forms of energy. The kinetic energy of a gas is due to the coordinated motion of all the molecules with the same average velocity in the same direction. The potential energy is due to the vantage position taken by the molecules or displacements of molecules from their normal positions. Heat or thermal energy is due to the random thermal motion of molecules in a completely disorderly fashion and the average velocity is zero. Orderly energy can be readily converted into disorderly energy, e.g. mechanical and electrical energies into internal energy (and then heat) by friction and Joule effect. Orderly energy can also be converted into one another. *But there are natural limitations on the conversion of disorderly energy into orderly energy, as delineated by the second law*. When work is dissipated into internal energy, the disorderly motion of molecules is increased. Two gases, when mixed, represent a higher degree of disorder than when they are separated. An irreversible process always tends to take the system (isolated) to a state of greater disorder. It is a tendency on the part of nature to proceed to a state of greater disorder. An isolated system always tends to a state of greater entropy. So there is a close link between entropy and disorder. It may be stated roughly that *the entropy of a system is a measure of the degree of molecular disorder existing in the system*. When heat is imparted to a system, the disorderly motion of molecules increases, and so the entropy of the system increases. The reverse occurs when heat is removed from the system.

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The entropy change of the working fluid in the refrigerator would be zero, since it is operating in a cycle, i.e

$$(\Delta S)_{ref} = 0$$

The entropy change of the atmosphere (positive)

$$(\Delta S)_{atm} = \frac{Q + W}{T}$$

∴ Entropy change of the universe

$$(\Delta S)_{univ} = (\Delta S)_{system} + (\Delta S)_{ref} + (\Delta S)_{atm}$$

$$= (S_1 - S_4) + \frac{Q + W}{T}$$

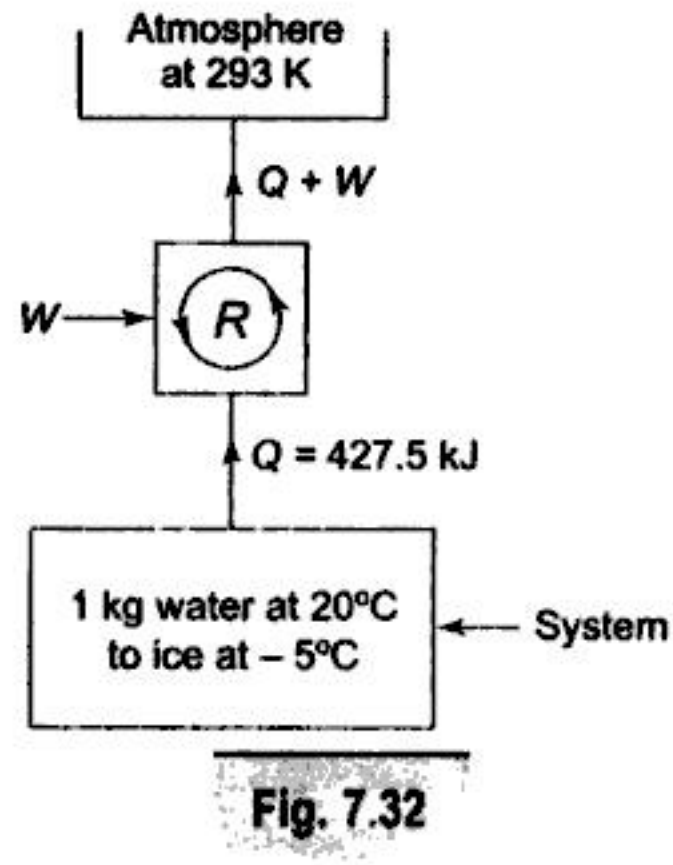

Fig. 7.32

By the principle of increase of entropy

$$(\Delta S)_{univ \text{ or isolated system}} \geq 0$$

$$\therefore \quad \left[(S_1 - S_4) + \frac{Q + W}{T}\right] \geq 0$$

$$\therefore \quad \frac{Q + W}{T} \geq (S_4 - S_1)$$

$$W \geq T(S_4 - S_1) - Q$$

$$\therefore \quad W(\text{min}) = T(S_4 - S_1) - Q$$

Here $Q = 427.5$ kJ

$T = 293$ K

$S_4 - S_1 = 1.5549$ kJ/K

$$\therefore \quad W(\text{min}) = 293 \times 1.5549 - 427.5 = \mathbf{28.5\ kJ}$$

Example 7.4 Two identical bodies of constant heat capacity are at the same initial temperature T_i. A refrigerator operates between these two bodies until one body is cooled to temperature T_2. If the bodies remain at constant pressure and undergo no change of phase, show that the minimum amount of work needed to do this is

$$W(\text{min}) = C_p \left(\frac{T_i^2}{T_2} + T_2 - 2T_i\right)$$

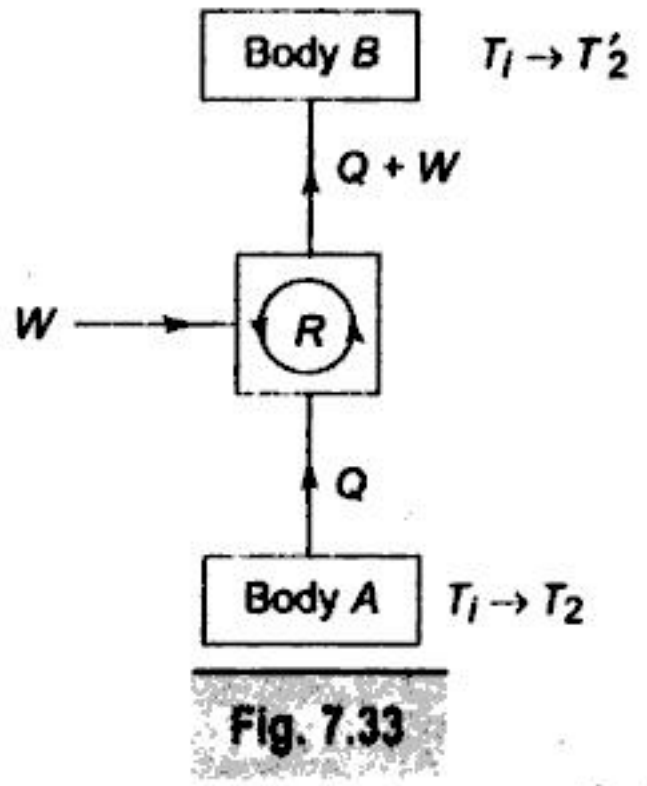

Fig. 7.33

Solution Both the finite bodies A and B are initially at the same temperature T_i. Body A is to be cooled to temperature T_2 by operating the refrigerator cycle, as shown in Fig. 7.33. Let T_2' be the final temperature of body B.

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$$(\Delta S)_{res} = \frac{Q_1 - W}{T_{res}} = \frac{98 \times 10^3 - W}{100} \text{ J/K}$$

$$(\Delta S)_{\text{working fluid in H.E.}} = 0$$

$$\therefore \quad (\Delta S)_{univ} = (\Delta S)_{system} + (\Delta S)_{res}$$

$$= -630 + \frac{98 \times 10^3 - W}{100}$$

Since $(\Delta S)_{univ} \geq 0$

$$\therefore \quad -630 + \frac{98 \times 10^3 - W}{100} \geq 0$$

$$980 - \frac{W}{100} - 630 \geq 0$$

$$\frac{W}{100} \leq 350$$

$$W(\text{max}) = 35{,}000 \text{ J} = \mathbf{35 \text{ kJ}}$$

Example 7.7 A fluid undergoes a reversible adiabatic compression from 0.5 MPa, 0.2 m^3 to 0.05 m^3 according to the law, $pv^{1.3}$ = constant. Determine the change in enthalpy, internal energy and entropy, and the heat transfer and work transfer during the process.

Solution

$$TdS = dH - Vdp$$

For the reversible adiabatic process (Fig. 7.36)

$$dH = Vdp$$

$$p_1 = 0.5 \text{ MPa}, \quad V_1 = 0.2 \text{ m}^3$$

$$V_2 = 0.05 \text{ m}^3, \quad p_1 V_1^n = p_2 V_2^n$$

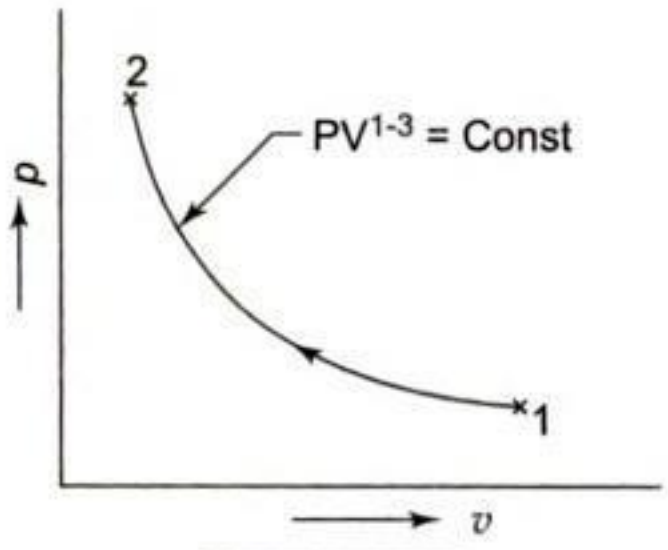

Fig. 7.36

$$\therefore \quad p_2 = p_1 \left(\frac{V_1}{V_2}\right)^n$$

$$= 0.5 \times \left(\frac{0.20}{0.05}\right)^{1.3} \text{ MPa}$$

$$= 0.5 \times \left(\frac{0.20}{0.05}\right)^{1.3} \text{ MPa}$$

$$= 0.5 \times 6.061 \text{ MPa} = 3.0305 \text{ MPa}$$

$$p_1 V_1^n = pV^n$$

$$\therefore \quad V = \left(\frac{p_1 V_1^n}{p}\right)^{1/n}$$

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$$0 = \Sigma \frac{\dot{Q}}{T} + \dot{S}_{\text{gen.total}}$$

$$0 = \frac{1164.84}{300} - \frac{1164.84}{275} + \dot{S}_{\text{gen.total}}$$

$$\dot{S}_{\text{gen.total}} = 4.236 - 3.883 = \mathbf{0.353\ W/K}$$

Review Questions

7.1 Show that through one point there can pass only one reversible adiabatic.
7.2 State and prove Clausius' theorem.
7.3 Show that entropy is a property of a system.
7.4 How is the entropy change of a reversible process estimated? Will it be different for an irreversible process between the same end states?
7.5 Why is the Carnot cycle on *T-s* plot a rectangle?
7.6 Establish the inequality of Clausius.
7.7 Give the criteria of reversibility, irreversibility and impossibility of a thermodynamic cycle.
7.8 What do you understand by the entropy principle?
7.9 When the system is at equilibrium, why would any conceivable change in entropy be zero?
7.10 Why is the entropy increase of an isolated system a measure of the extent of irreversibility of the process undergone by the system?
7.11 How did Rudolf Clausius summarize the first and second laws of thermodynamics?
7.12 Show that the transfer of heat through a finite temperature difference is irreversible.
7.13 Show that the adiabatic mixing of two fluids is irreversible.
7.14 What is the maximum work obtainable from two finite bodies at temperatures T_1 and T_2?
7.15 Determine the maximum work obtainable by using one finite body at temperature T and a thermal energy reservoir at temperature T_0, $T > T_0$.
7.16 What are the causes of entropy increase?
7.17 Why is an isentropic process not necessarily an adiabatic process?
7.18 What is the reversible adiabatic work for a steady flow system when K.E. and P.E. changes ae negligibly small? How is it different from that for a closed stationary system?
7.19 Why are the equations

$$TdS = dU + pdV$$

$$TdS = dH - Vdp$$

valid for any process between two equilibrium end states?
7.20 Why is the second law called a directional law of nature?
7.21 How is entropy related to molecular disorder in a system?
7.22 Show that entropy varies logarithmically with the disorder number.

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$$\frac{c_p}{R} = a + bT + cT^2, \text{ where } a = 3.634,$$

$$b = 1.195 \times 10^{-3}\ \text{K}^{-1} \text{ and } c = 0.135 \times 10^{-6}\ \text{K}^{-2}$$

Calculate the entropy change of the system.

Ans. 277.8 J/K

7.33 A 50 ohm resistor carrying a constant current of 1 A is kept at a constant temperature of 27°C by a stream of cooling water. In a time interval of 1 s, (a) what is the change in entropy of the resistor? (b) What is the change in entropy of the universe?

Ans. (a) 0, (b) 0.167 J/K

7.34 A lump of ice with a mass of 1.5 kg at an initial temperature of 260 K melts at the pressure of 1 bar as a result of heat transfer from the environment. After some time has elapsed the resulting water attains the temperature of the environment, 293 K. Calculate the entropy production associated with this process. The latent heat of fusion of ice is 333.4 kJ/kg, the specific heat of ice and water are 2.07 and 4.2 kJ/kg K respectively, and ice melts at 273.15 K.

Ans. 0.1514 kJ/K

7.35 An ideal gas is compressed reversibly and adiabatically from state a to state b. It is then heated reversibly at constant volume to state c. After expanding reversibly and adiabatically to state d such that $T_b = T_d$, the gas is again reversibly heated at constant pressure to state e such that $T_e = T_c$. Heat is then rejected reversibly from the gas at constant volume till it returns to state a. Express T_a in terms of T_b and T_c. If $T_b = 555$ K and $T_c = 835$ K, estimate T_a. Take $\gamma = 1.4$.

Ans. $T_a = \dfrac{T_b^{\gamma+1}}{T_c^{\gamma}}$, 313.29 K

7.36 Liquid water of mass 10 kg and temperature 20°C is mixed with 2 kg of ice at – 5°C till equilibrium is reached at 1 atm pressure. Find the entropy change of the system. Given: c_p of water = 4.18 kJ/kg K, c_p of ice = 2.09 kJ/kg K and latent heat of fusion of ice = 334 kJ/kg.

Ans. 190 J/K

7.37 A thermally insulated 50-ohm resistor carries a current of 1 A for 1 s. The initial temperature of the resistor is 10°C. Its mass is 5 g and its specific heat is 0.85 J/g K. (a) What is the change in entropy of the resistor? (b) What is the change in entropy of the universe?

Ans. (a) 0.173 J/K (b) 0.173 J/K

7.38 The value of c_p for a certain substance can be represented by $c_p = a + bT$. (a) Determine the heat absorbed and the increase in entropy of a mass m of the substance when its temperature is increased at constant pressure from T_1 to T_2. (b) Find the increase in the molal specific entropy of copper, when the temperature is increased at constant pressure from 500 to 1200 K. Given for copper: when $T = 500$ K, $c_p = 25.2 \times 10^3$ and when $T = 1200$ K, $c_p = 30.1 \times 10^3$ J/k mol K.

Ans. (a) $m\left[a(T_2 - T_1) + \dfrac{b}{2}\left(T_2^2 - T_1^2\right), m\left[a \ln \dfrac{T_2}{T_1} + b(T_2 - T_2)\right]\right]$,

(b) 24.7 kJ/k mol K

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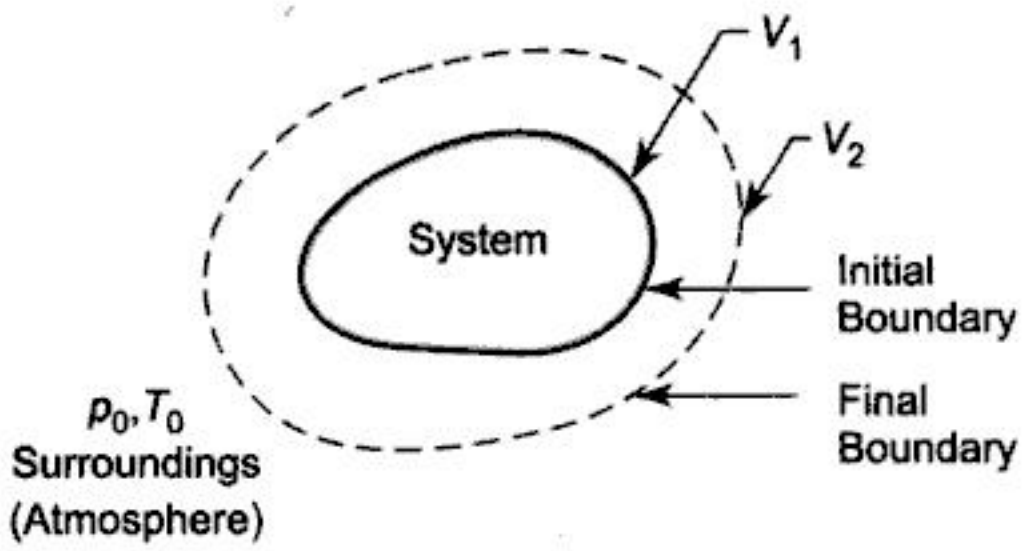

Fig. 8.12 *Work Done by a Closed System in Pushing Out the Atmosphere*

Similarly, the *maximum useful work* will be

$$(W_u)_{\max} = W_{\max} - p_0 (V_2 - V_1) \tag{8.35}$$

In differential form

$$(dW_u)_{\max} = dW_{\max} - p_0\, dV \tag{8.36}$$

In a *steady flow system*, the volume of the system does not change. Hence, the maximum useful work would remain the same, i.e. *no work is done on the atmosphere*, or

$$(đ W_u)_{\max} = đ W_{\max} \tag{8.37}$$

But in the case of an unsteady-flow open system or a closed system, the volume of the system changes. Hence, when a system exchanges heat only with the atmosphere, the maximum useful work becomes

$$(đ W_u)_{\max} = đ W_{\max} - p_0\, dV$$

Substituting $dW_{\max}$ from Eq. (8.24),

$$(đ W_u) = dm_1 \left(h_1 - T_0 s_1 + \frac{\mathbf{V}_1^2}{2} + g z_1 \right) - dm_2 \left(h_2 - T_0 s_2 + \frac{\mathbf{V}_2^2}{2} + g z_2 \right)$$

$$- d \left[U + p_0 V - T_0 S + \frac{m\mathbf{V}^2}{2} + mgz \right]_\sigma \tag{8.38}$$

This is the maximum useful work for an *unsteady open system*. For the *closed system* Eq. (8.38) reduces to

$$(đ W_u)_{\max} = -d \left[U + p_0 V - T_0 S + \frac{m\mathbf{V}^2}{2} + mgz \right]_\sigma$$

$$= -d[E + p_0\, V - T_0\, S]_\sigma \tag{8.39}$$

$$\therefore \quad (W_u)_{\max} = E_1 - E_2 + p_0 (V_1 - V_2) - T_0 (S_1 - S_2) \tag{8.40}$$

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8.8.2 Availability in a Nonflow Process

Let us consider a closed system and denote its initial state by parameters without any subscript and the final dead state with subscript '0'. The availability of the system A, i.e. the maximum useful work obtainable as the system reaches the dead state, is given by Eq. (8.40)

$$A = (W_u)_{\max} = E - E_0 + p_0 (V - V_0) - T_0(S - S_0)$$

$$= \left(U + \frac{m\mathbf{V}^2}{2} + mgz \right) - (U_0 + mgz_0) + p_0(V - V_0) - T_0(S - S_0) \quad (8.51)$$

If K.E. and P.E. changes are neglected and for unit mass, the availability becomes

$$a = u - u_0 + p_0 (v - v_0) - T_0 (s - s_0)$$
$$= (u + p_0 v - T_0 s) - (u_0 + p_0 v_0 - T_0 s_0)$$
$$= \phi - \phi_0 \quad (8.52)$$

where ϕ is the availability function of the closed system.

If the system undergoes a change of state from 1 to 2, the decrease in availability will be

$$a = (\phi_1 - \phi_0) - (\phi_2 - \phi_0)$$
$$= \phi_1 - \phi_2$$
$$= (u_1 - u_2) + p_0 (v_1 - v_2) - T_0(s_1 - s_2) \quad (8.53)$$

This is the maximum useful work obtainable under the given surroundings.

8.9 AVAILABILITY IN CHEMICAL REACTIONS

In many chemical reactions the reactants are often in pressure and temperature equilibrium with the surroundings (before the reaction takes place) and so are the products after the reaction. An internal combustion engine can be cited as an example of such a process if we visualize the products being cooled to atmospheric temperature T_0 before being discharged from the engine.

(a) Let us first consider a system which is in temperature equilibrium with the surroundings before and after the process. The maximum work obtainable during a change of state is given by Eq. (8.31),

$$W_{\max} = E_1 - E_2 - T_0(S_1 - S_2)$$

$$= \left(U_1 + \frac{m\mathbf{V}_1^2}{2} + mgz_1 \right) - \left(U_2 + \frac{m\mathbf{V}_2^2}{2} + mgz_2 \right) - T_0(S_1 - S_2)$$

If K.E. and P.E. changes are neglected,

$$W_{\max} = U_1 - U_2 - T_0 (S_1 - S_2)$$

Since the initial and final temperatures of the system are the same as that of the surroundings, $T_1 = T_2 = T_0 = T$, say, then

$$(W_T)_{\max} = (U_1 - U_2)_T - T(S_1 - S_2)_T \quad (8.54)$$

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Here, $\dot{m}_1 + \dot{m}_2 = \dot{m}_3 = \dot{m}$(say)

Let $$x = \frac{\dot{m}_1}{\dot{m}_1 + \dot{m}_2}$$

By the first law,

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

or $$x h_1 + (1 - x) h_2 = h_3$$

The preceding equation may be written in the following form, since enthalpy is a function of temperature.

$$x T_1 + (1 - x) T_2 = T_3$$

$$\therefore \quad \frac{T_3}{T_1} = x + (1 - x)\tau \tag{8.70}$$

where $$\tau = \frac{T_2}{T_1}$$

By the second law,

$$\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

$$= \dot{m} s_3 - x \dot{m} s_1 - (1 - x)\ \dot{m} s_2$$

$$\therefore \quad \frac{\dot{S}_{gen}}{\dot{m}} = (s_3 - s_2) + x(s_2 - s_1)$$

$$= c_p \ln \frac{T_3}{T_2} + x c_p \ln \frac{T_2}{T_1}$$

or $$\frac{\dot{S}_{gen}}{\dot{m} c_p} = \ln \left(\frac{T_3}{T_2} \right) \left(\frac{T_2}{T_1} \right)^x$$

or $$N_s = \ln \left(\frac{T_3}{T_1} \right) \frac{T_1^{1-x}}{T_2^{1-x}}$$

or $$N_s = \ln \frac{T_3 / T_1}{(T_2 / T_1)^{1-x}} \tag{8.71}$$

where N_S is a dimensionless quantity, called the *entropy generation number*, given by $\dot{S}_{gen} / \dot{m} c_p$.

Substituting T_3/T_1 from Eq. (8.70) in Eq. (8.71),

$$Ns = \ln \frac{x + \tau (1 - x)}{\tau^{1-x}} \tag{8.72}$$

If $x = 1$ or $\tau = 1$, N_s becomes zero in each case. The magnitude of N_s depends on x and τ. The rate of loss of exergy due to mixing would be

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$$= \frac{W - W'}{W} = \frac{T_0 (\Delta S' - \Delta S)}{(T_1 - T_0)\,\Delta S} = \frac{T_0 \left(\dfrac{\Delta S'}{\Delta S} - 1 \right)}{(T_1 - T_0)}$$

$$= \frac{T_0 (T_1 - T_1')}{T_1' (T_1 - T_0)} = \frac{308 (693 - 523)}{523 (693 - 308)} = \mathbf{0.26}$$

Example 8.2 In a steam boiler, hot gases from a fire transfer heat to water which vaporizes at constant temperature. In a certain case, the gases are cooled from 1100 °C to 550°C while the water evaporates at 220°C. The specific heat of gases is 1.005 kJ/kg-K, and the latent heat of water at 220°C, is 1858.5 kJ/kg. All the heat transferred from the gases goes to the water. How much does the total entropy of the combined system of gas and water increase as a result of the irreversible heat transfer? Obtain the result on the basis of 1 kg of water evaporated.

If the temperature of the surroundings is 30°C, find the increase in unavailable energy due to irreversible heat transfer.

Solution Gas ($\dot{m}_g$) is cooled from state 1 to state 2 (Fig. 8.25). For reversible heat transfer, the working fluid (w.f.) in the heat engine having the same c_p would have been heated along 2 – 1, so that at any instant, the temperature difference between gas and the working fluid is zero. Then 1 – b would have been the expansion of the working fluid down to the lowest possible temperature T_0, and the amount of heat rejection would have been given by the area *abcd*.

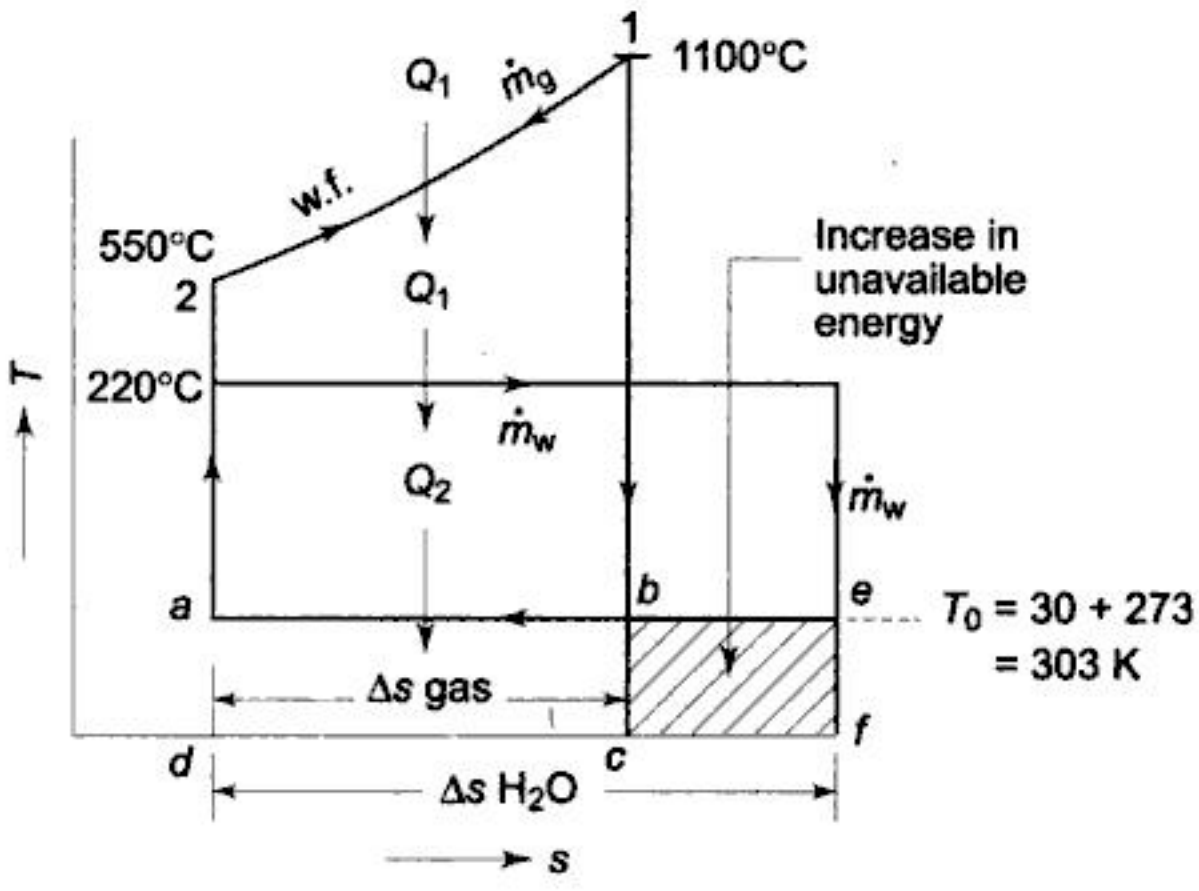

Fig. 8.25

When water evaporates at 220°C as the gas gets cooled from 1100°C to 550°C, the resulting power cycle has an unavailable energy represented by the area *aefd*. The increase in unavailable energy due to irreversible heat transfer is thus given by area *befc*.

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Example 8.13 A system undergoes a power cycle while receiving energy Q_1 at temperature T_1 and discharging energy Q_2 at temperature T_2. There are no other heat transfers.

(a) Show that the thermal efficiency of the cycle can be expressed as:

$$\eta = 1 - \frac{T_2}{T_1} - \frac{T_2 I}{T_0 Q_1}$$

where T_0 is the ambient temperature and I is the irreversibility of the cycle.

(b) Obtain an expression for the maximum theoretical value for the thermal efficiency.

(c) Derive an expression for the irreversibility for which no network is developed by the cycle. What conclusion do you derive from it?

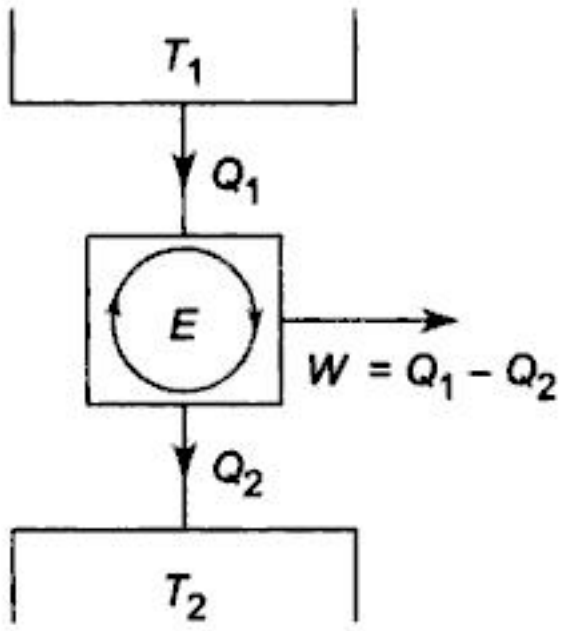

Fig. 8.28

Solution An availability balance for the cycle gives (Fig. 8.28)

$$(\Delta A)_{\text{cycle}} = 0 = \left(1 - \frac{T_0}{T_1}\right) Q_1 - \left(1 - \frac{T_0}{T_2}\right) Q_2 - W - I$$

since each property is restored to its initial state.

Since $Q_2 = Q_1 - W$,

$$0 = \left(1 - \frac{T_0}{T_1}\right) Q_1 - \left(1 - \frac{T_0}{T_2}\right)(Q_1 - W) - W - I$$

$$W = \frac{T_2}{T_0}\left[\left(1 - \frac{T_0}{T_1}\right) - \left(1 - \frac{T_0}{T_2}\right)\right] Q_1 - \frac{T_2 I}{T_0}$$

$$\eta = \frac{W}{Q_1} = 1 - \frac{T_2}{T_1} - \frac{T_2 I}{T_0 Q_1} \qquad \textit{Proved.}$$

(b) When $I = 0$,

$$\eta_{\max} = 1 - \frac{T_2}{T_1}$$

(c) When $W = 0$

$$\eta = 0 = 1 - \frac{T_2}{T_1} - \frac{T_2 I}{T_0 Q_1}$$

$$I = T_0\left[\frac{1}{T_2} - \frac{1}{T_1}\right] Q_1 = T_0\left[\frac{Q_1}{T_2} - \frac{Q_1}{T_1}\right] = T_0 S_{\text{gen}}$$

The heat transfer Q_1 from T_1 to T_2 takes place through a reversible engine, and the entire work is dissipated in the brake, from which an equal amount of heat is

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8.43 How can you improve the first law efficiency and the second law efficiency?

Problems

8.1 What is the maximum useful work which can be obtained when 100 kJ are abstracted from a heat reservoir at 675 K in an environment at 288 K? What is the loss of useful work if (a) a temperature drop of 50°C is introduced between the heat source and the heat engine, on the one hand, and the heat engine and the heat sink, on the other, (b) the source temperature drops by 50°C and the sink temperature rises by 50°C during the heat transfer process according to the linear law $\frac{dQ}{dT} = \pm$ constant? Ans. (a) 11.2 kJ, (b) 5.25 kJ

8.2 In a steam generator, water is evaporated at 260°C, while the combustion gas (c_p = 1.08 kJ/kg K) is cooled from 1300°C to 320°C. The surroundings are at 30°C. Determine the loss in available energy due to the above heat transfer per kg of water evaporated. (Latent heat of vaporization of water at 260°C = 1662.5 kJ/kg.) Ans. 443.6 kJ

8.3 Exhaust gases leave an internal combustion engine at 800°C and 1 atm, after having done 1050 kJ of work per kg of gas in the engine (c_p of gas = 1.1 kJ/kg K). The temperature of the surroundings is 30°C. (a) How much available energy per kg of gas is lost by throwing away the exhaust gases? (b) What is the ratio of the lost available energy to the engine work?
Ans. (a) 425.58 kJ, (b) 0.405

8.4 A hot spring produces water at a temperature of 56°C. The water flows into a large lake, with a mean temperature of 14°C, at a rate of 0.1 m^3 of water per min. What is the rate of working of an ideal heat engine which uses all the available energy? Ans. 19.5 kW

8.5 0.2 kg of air at 300°C is heated reversibly at constant pressure to 2066 K. Find the available and unavailable energies of the heat added. Take T_0 = 30°C and c_p = 1.0047 kJ/kg K. Ans. 211.9 and 78.1 kJ

8.6 Eighty kg of water at 100°C are mixed with 50 kg of water at 60°C, while the temperature of the surroundings is 15°C. Determine the decrease in available energy due to mixing. Ans. 236 kJ

8.7 A lead storage battery used in an automobile is able to deliver 5.2 MJ of electrical energy. This energy is available for starting the car.
Let compressed air be considered for doing an equivalent amount of work in starting the car. The compressed air is to be stored at 7 MPa, 25°C. What is the volume of the tank that would be required to let the compressed air have an availability of 5.2 MJ? For air, $pv = 0.287\ T$, where T is in K, p in kPa, and v in m^3/kg. Ans. 0.228 m^3

8.8 Ice is to be made from water supplied at 15°C by the process shown in Fig. 8.33. The final temperature of the ice is – 10°C, and the final temperature of the water that is used as cooling water in the condenser is 30°C. Determine the minimum work required to produce 1000 kg of ice. Ans. 33.37 MJ
Take c_p for water = 4.187 kJ/kg K, c_p for ice = 2.093 kJ/kg K, and latent heat of fusion of ice = 334 kJ/kg.

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8.33 In a counterflow heat exchanger, oil (c_p = 2.1 kJ/kg-K) is cooled from 440 to 320 K, while water (c_p = 4.2 kJ/kg K) is heated from 290 K to temperature T. The respective mass flow rates of oil and water are 800 and 3200 kg/h. Neglecting pressure drop, KE and PE effects and heat loss, determine (a) the temperature T, (b) the rate of exergy destruction, (c) the second law efficiency. Take T_0 = 17°C and p_0 = 1 atm. Ans. (a) 305 K, (b) 41.4 MJ/h, (c) 10.9%

8.34 Oxygen enters a nozzle operating at steady state at 3.8 MPa, 387°C and 10 m/s. At the nozzle exit the conditions are 150 kPa, 37°C and 750 m/s. Determine (a) the heat transfer per kg and (b) the irreversibility. Assume oxygen as an ideal gas, and take T_0 = 20°C, p_0 = 1 atm.

Ans. (a) – 37.06 kJ/kg, (b) 81.72 kJ/kg

8.35 Argon gas expands adiabatically in a turbine from 2 MPa, 1000°C to 350 kPa. The mass flow rate is 0.5 kg/s and the turbine develops power at the rate of 120 kW. Determine (a) the temperature of argon at the turbine exit, (b) the irreversibility rate, and (c) the second law efficiency. Neglect KE and PE effects and take T_0 = 20°C, p_0 = 1 atm. Ans. (a) 538.1°C, b) 18.78 kW, (c) 86.5%

8.36 In the boiler of a power plant are tubes through which water flows as it is brought from 0.8 MPa, 150°C (h = 632.6 kJ/kg, s = 1.8418 kJ/kg K) to 0.8 MPa, 250°C (h = 2950 kJ/kg, s = 7.0384 kJ/kg K). Combustion gases passing over the tubes cool from 1067°C to 547°C. These gases may be considered as air (ideal gas) having c_p = 1.005 kJ/kg K. Assuming steady state and neglecting any heat loss, and KE and PE effects, determine (a) the mass flow rate of combustion gases per kg of steam, (b) the loss of exergy per kg steam, and (c) the second law efficiency. Take T_0 = 25°C, p_0 = 1 atm.

Ans. (a) m_g/m_w = 4.434, (b) 802.29 kJ/kg steam, (c) 48.9%

8.37 Air enters a hair dryer at 22°C, 1 bar with a velocity of 3.7 m/s and exits at 83°C, 1 bar with a velocity of 9.1 m/s through an area of 18.7 cm^2. Neglecting any heat loss and PE effect and taking T_0 = 22°C, (a) evaluate the power required in kW, and (b) devise and evaluate a second law efficiency. Ans. (a) – 1.02 kW, (b) 9%

8.38 An isolated system consists of two solid blocks. One block has a mass of 5 kg and is initially at 300°C. The other block has a mass of 10 kg and is initially at – 50°C. The blocks are allowed to come into thermal equilibrium. Assuming the blocks are incompressible with constant specific heats of 1 and 0.4 kJ/kg K, respectively, determine (a) the final temperature, (b) the irreversibility. Take T_0 = 300 K. Ans. (a) 417.4 K, (b) 277 kJ

8.39 Air flows into a heat engine at ambient conditions 100 kPa, 300 K. Energy is supplied as 1200 kJ per kg air from a 1500 K source and in some part of the process, a heat loss of 300 kJ/kg air happens at 750 K. The air leaves the engine at 100 kPa, 800 K. Find the first and the second law efficiencies.

Ans. 0.315, 0.672

8.40 Consider two rigid containers each of volume 1 m^3 containing air at 100 kPa, 400 K. An internally reversible Carnot heat pump is then thermally connected between them so that it heats one up and cools the other down. In order to transfer heat at a reasonable rate, the temperature difference between the working fluid inside the heat pump and the air in the containers is set to 20°C. The process stops when the air in the coldest tank reaches 300 K. Find the final temperature of the air that is heated up, the work input to the heat pump, and the overall second law efficiency. Ans. 550 K, 31.2 kJ, 0.816

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1. The entropy increase of ice as it is heated from – 5°C to 0°C at 1 atm. ($c_{\text{pice}} = 2.093$ kJ/kg K).

$$\Delta s_1 = s_2 - s_1 = \int \frac{đQ}{T} = \int_{T_1 = 268}^{T_2 = 273} \frac{mc_p dT}{T} = mc_p \ln \frac{273}{268}$$

$$= 1 \times 2.093 \ln \frac{273}{268} = 0.0398 \text{ kJ/kg K}$$

2. The entropy increase of ice as it melts into water at 0°C (latent heat of fusion of ice = 334.96 kJ/kg)

$$\Delta s_2 = s_3 - s_2 = \frac{334.96}{273} = 1.23 \text{ kJ/kg K}$$

3. The entropy increase of water as it is heated from 0°C to 100°C ($c_{p_{\text{water}}} = 4.187$ kJ/kg K)

$$\Delta s_3 = s_4 - s_3 = mc_p \ln \frac{T_3}{T_2} = 1 \times 4.187 \ln \frac{373}{273} = 1.305 \text{ kJ/kg K}$$

4. The entropy increase of water as it is vaporized at 100°C, absorbing the latent heat of vaporization (2257 kJ/kg)

$$\Delta s_4 = s_5 - s_4 = \frac{2257}{273} = 6.05 \text{ kJ/kg K}$$

5. The entropy increase of vapour as it is heated from 100°C to 250°C at 1 atm

$$\Delta s_5 = s_6 - s_5 = \int_{373}^{523} mc_p \frac{dT}{T} = 1 \times 2.093 \ln \frac{523}{373}$$

$$= 0.706 \text{ kJ/kg K}$$

assuming the average specific heat of steam in the temperature range of 100°C to 250°C as 2.093 kJ/kg K.

These entropy changes are shown in Fig. 9.11. The curve 1-2-3-4-5-6 is the isobar of 1 atm. If, during the heating process, the pressure had been maintained constant at 2 atm, a similar curve would be obtained. The states 2, 3, 4, and 5 are saturation states. If these states for different pressures are joined, as in Figs. 9.3 and 9.4, the phase equilibrium diagram of a pure substance on the *T*-*s* coordinates, as shown in Fig. 9.12, would be obtained.

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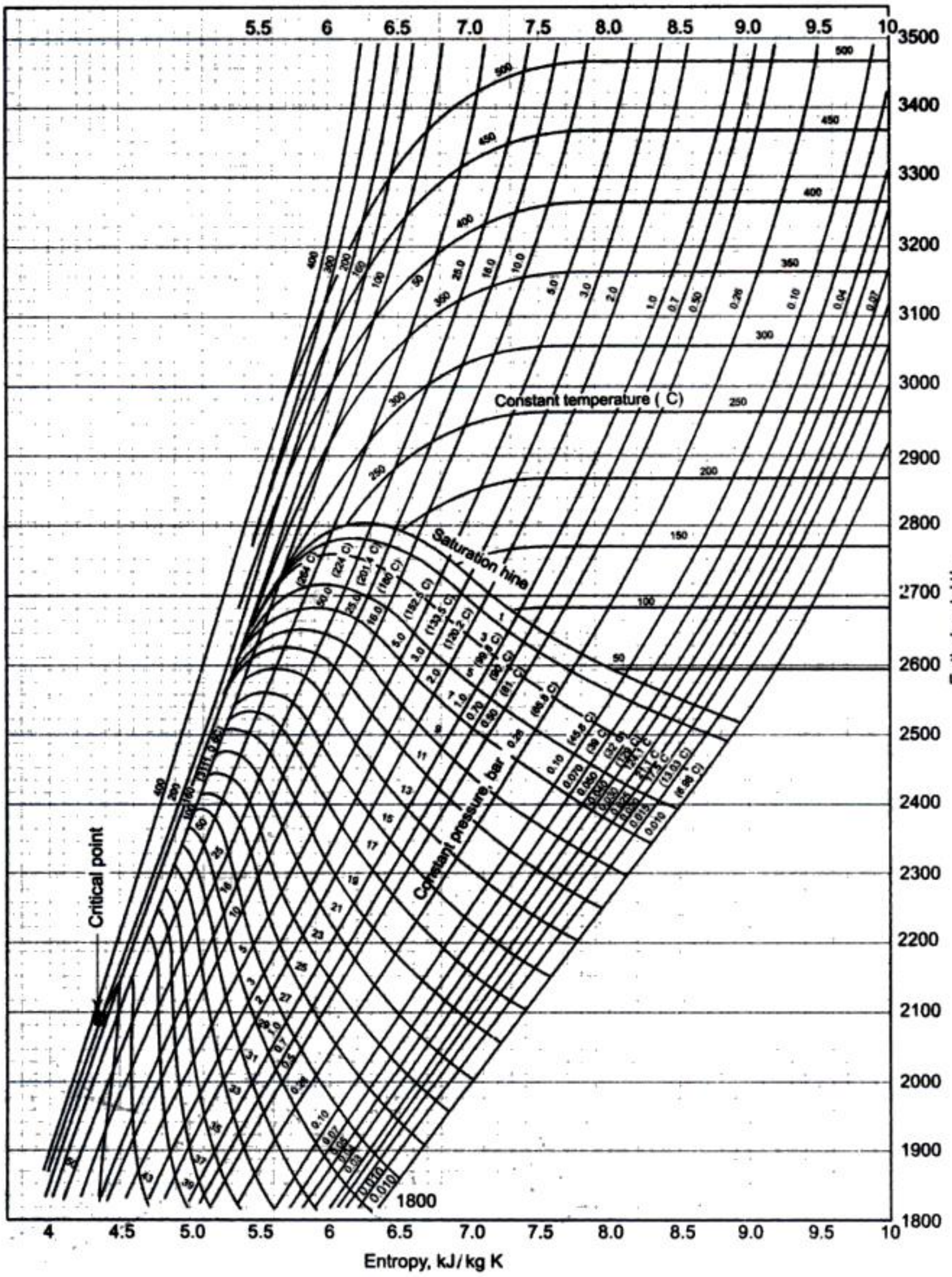

Fig. 9.21 *Mollier Diagram for Steam (Data Taken From Keenan, J.H., F.G. Keyes. P.C. Hill and J.G. Moore, Steam Tables, John Wiley, N.Y., 1969)*

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Therefore, x_2, the quality of steam after partial moisture separation, can be evaluated. If m kg of steam is taken through the sampling tube in t secs, m_1 kg of it is separated, and m_2 kg is throttled and then condensed to water and collected, then $m = m_1 + m_2$, and at state 2, the mass of dry vapour will be $x_2 m_2$. Therefore, the quality of the sample of steam at state 1, x_1 is given by

$$x_1 = \frac{\text{mass of dry vapour at state 1}}{\text{mass of liquid} - \text{vapour mixture at state 1}}$$

$$= \frac{x_2 m_2}{m_1 + m_2}$$

The quality of wet steam can also be measured by an *electric calorimeter* (Fig. 9.28). The sample of steam is passed in steady flow through an electric heater, as shown. The electrical energy input Q should be sufficient to take the steam to the superheated region where pressure p_2 and temperature t_2 are measured. If I is the current flowing through the heater in amperes and V is the voltage across the coil, then at steady state $Q = VI \times 10^{-3}$ kW. If m is the mass of steam taken in t seconds under steady flow condition, then the steady flow energy equation for the heater (as control volume) gives

$$w_1 h_1 + Q = w_1 h_2$$

where w_1 is the steam flow rate in kg/s $\left(w_1 = \frac{m}{t} \text{ kg/s} \right)$

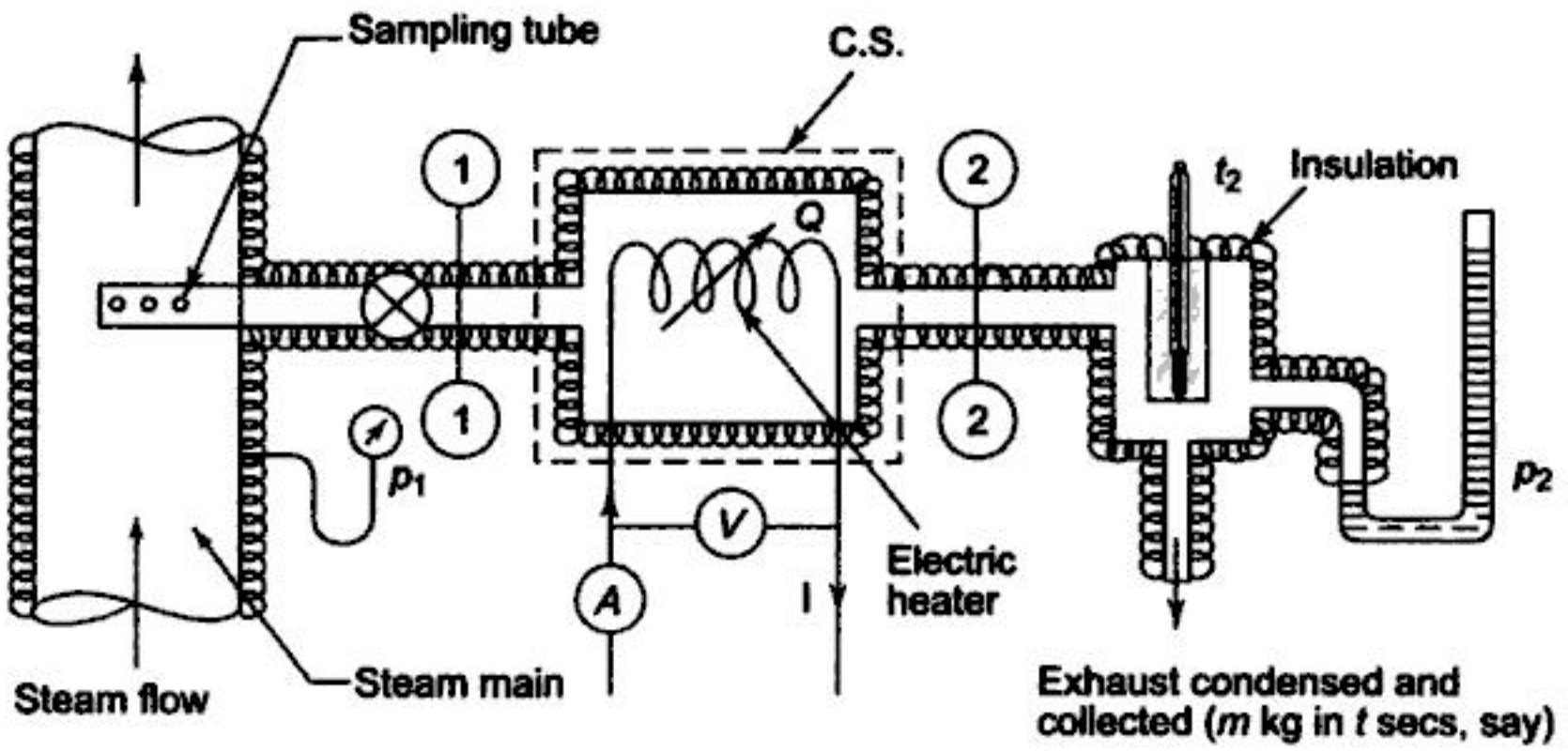

Fig. 9.28 *Electrical Calorimeter*

$$\therefore \qquad h_1 + \frac{Q}{w_1} = h_2$$

With h_2, Q and w_1 being known, h_1 can be computed. Now

$$h_1 = h_{fp1} + x_1 h_{fgp1}$$

Hence x_1 can be evaluated.

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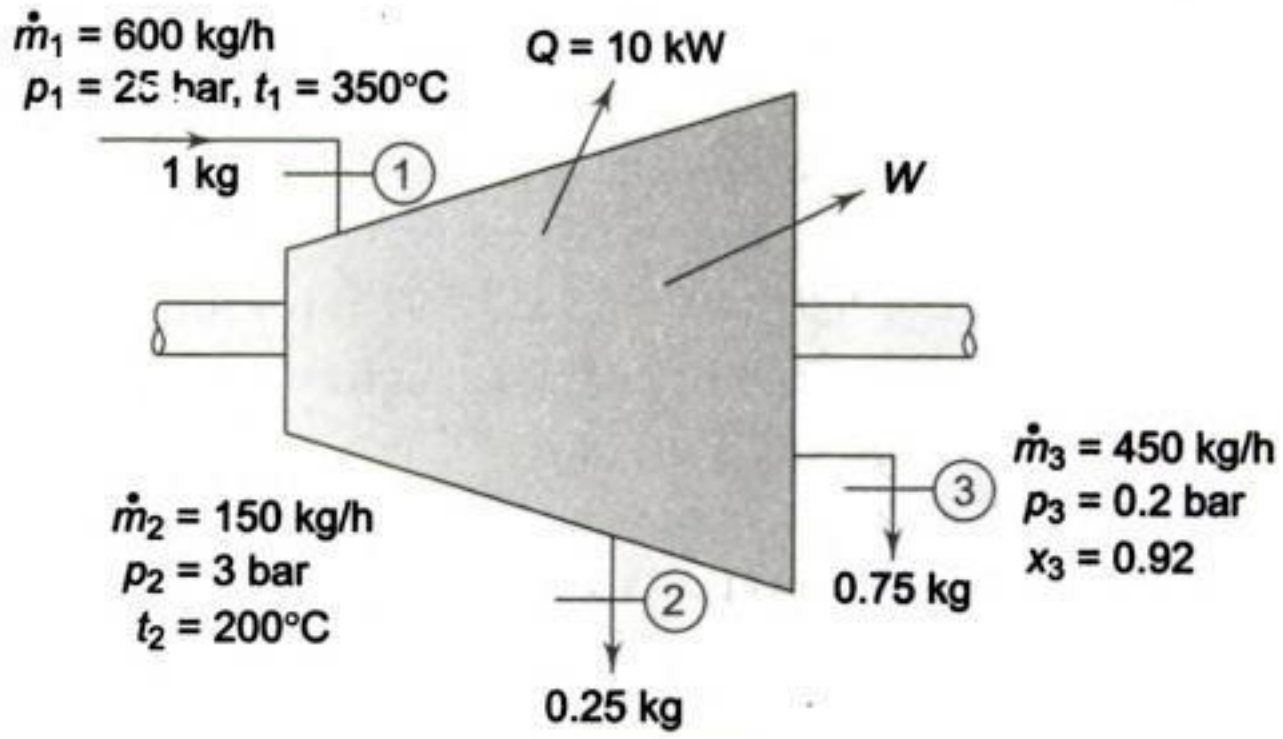

Fig. 9.41

Availability of steam leaving the turbine at state 3,

$$\Psi_3 = (h_3 - h_0) - T_0 (s_3 - s_0) = (2421.04 - 125.79) - 303 (7.3524 - 0.4369)$$
$$= 199.85 \text{ kJ/kg} \quad \text{Ans.}$$

(b) Maximum work per kg of steam entering the turbine

$$W_{rev} = \Psi_1 - \frac{m_2}{m_1} \Psi_2 - \frac{m_3}{m_1} \Psi_3 = 1057.48 - 0.25 \times 656.71 - 0.75 \times 199.85$$
$$= \mathbf{743.41 \text{ kJ/kg}}$$

(c) Irreversibility

$$I = T_0(w_2 s_2 + w_3 s_3 - w_1 s_1) - Q$$
$$= 303 (150 \times 7.3115 + 450 \times 7.3424 - 600 \times 6.8481) - (-10 \times 3600)$$
$$= 303 (1096.73 + 3304.08 - 4108.86) + 36000$$
$$= 124{,}460.85 \text{ kJ/h}$$
$$= 124.461 \text{ MJ/h} = \frac{124.461 \times 103}{600} = \mathbf{207.44 \text{ kJ/kg}}$$

Example 9.16 Determine the exergy of (a) 3 kg of water at 1 bar and 90°C, (b) 0.2 kg of steam at 4 MPa, 500°C and (c) 0.4 kg of wet steam at 0.1 bar and 0.85 quality, (d) 3 kg of ice at 1 bar – 10°C. Assume a dead state of 1 bar and 300 K.

Solution At the dead state of 1 bar, 300 K,

$$u_0 = 113.1 \text{ kJ/kg}, \quad h_0 = 113.2 \text{ kJ/kg K}$$
$$v_0 = 0.001005 \text{ m}^3\text{/kg}, \quad s_0 = 0.395 \text{ kJ/kg}$$

Exergy of the system:

$$\phi = m[(u + p_0 v - T_0 s) - (u_0 + p_0 v_0 - T_0 s_0)]$$

Now, $u_0 + p_0 v_0 - T_0 s_0 = h_0 - T_0 s_0$
$$= 113.2 - 300 \times 0.395 = -5.3 \text{ kJ/kg}$$

(a) For water at 1 bar, 90°C

$$u = 376.9 \text{ kJ/kg}, \quad h = 377 \text{ kJ/kg}, \quad v = 0.001035 \text{ m}^3\text{/kg}$$
$$s = 1.193 \text{ kJ/kg K}$$

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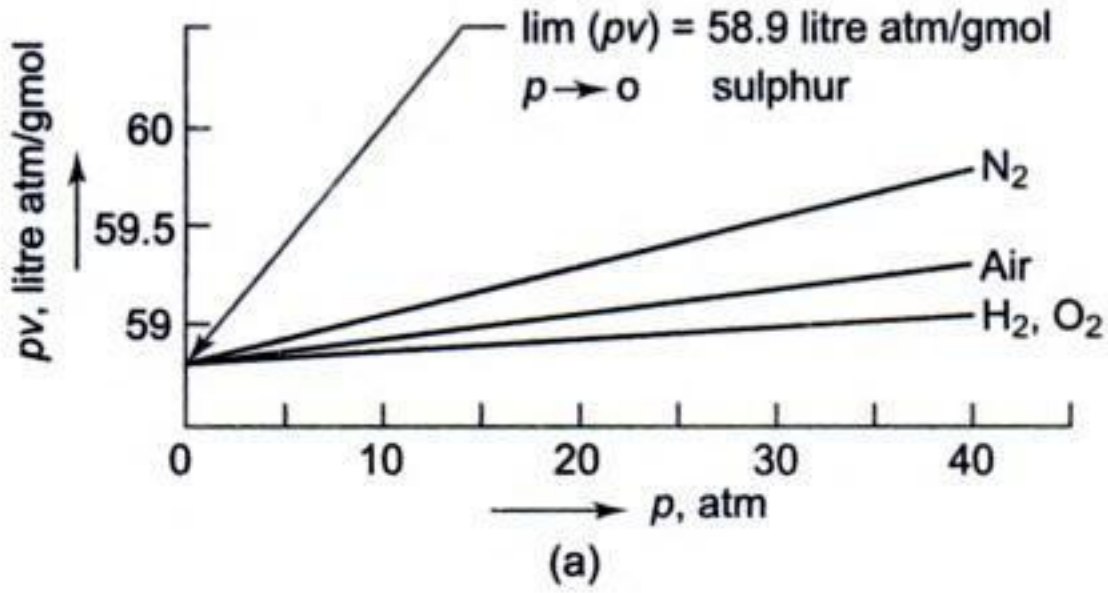

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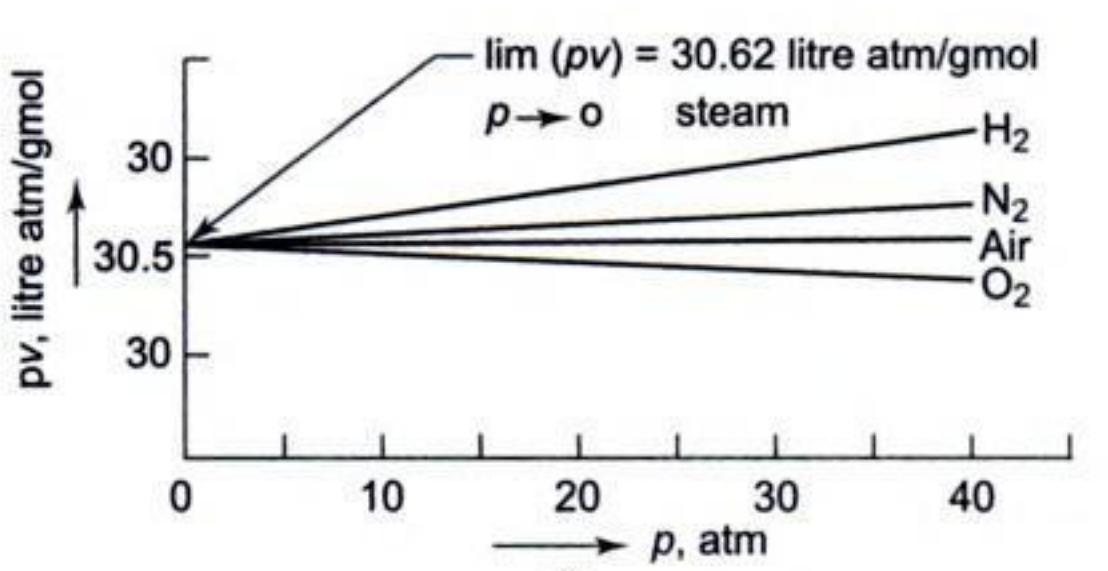

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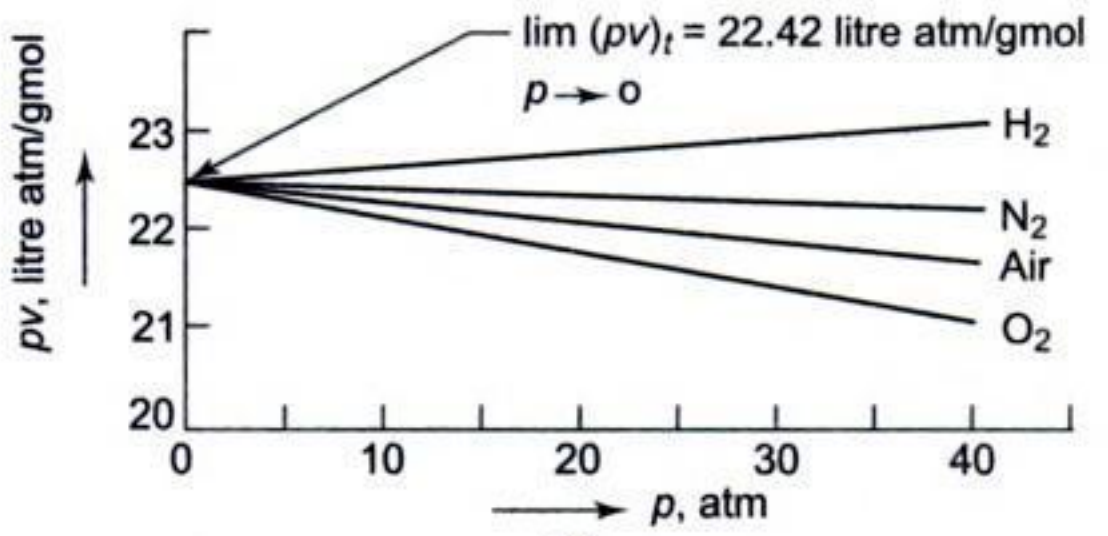

(c)

Fig. 10.1 *For Any, gas* $\lim_{p\to 0} (pV)_T$ *is Independent of the Nature of the Gas and Depends Only on T*

The value obtained for $\lim_{p\to 0} (pv)_t$ is $22.4 \frac{\text{litre} - \text{atm}}{\text{g mol}}$

$$\therefore \qquad \bar{R} = \frac{22.4}{273.16} = 0.083 \frac{\text{litre} - \text{atm}}{\text{g mol K}}$$

The equation of state of a gas is thus

$$\lim_{p\to 0} p\bar{v} = \bar{R}T \tag{10.3}$$

where v is the molar volume.

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For an ideal gas

$$pv = RT$$

From Eq. (10.33)

$$p = c \cdot v^{-\gamma}$$

$$\therefore \quad c \cdot v^{-\gamma} \cdot v = RT$$

$$c \cdot v^{1-\gamma} = RT$$

$$Tv^{\gamma-1} = \text{constant} \tag{10.34}$$

Between any two states 1 and 2, for a reversible adiabatic process in the case of an ideal gas

$$T_1 v_1^{\gamma-1} = T_2 v_2^{\gamma-1}$$

or

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} \tag{10.35}$$

Similarly, substituting from Eq. (10.33)

$$v = \left(\frac{c}{p}\right)^{1/\gamma} \text{ in the Eq. } pv = RT$$

$$p \cdot \frac{c'}{p^{1/\gamma}} = RT$$

$$p^{1-(1/\gamma)} \cdot c' = RT$$

$$\therefore \quad Tp^{(1-\gamma)/\gamma} = \text{constant} \tag{10.36}$$

Between any two states 1 and 2

$$T_1 p_1^{(1-\gamma)/\gamma} = T_2 p_2^{(1-\gamma)/\gamma}$$

$$\therefore \quad \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} \tag{10.37}$$

Equations (10.33), (10.34), and (10.36) give the relations among p, v, and T in a reversible adiabatic process for an ideal gas.

The *internal energy change* of an ideal gas for a reversible adiabatic process is given by

$$Tds = du + pdv = 0$$

or

$$\int_1^2 du = -\int_1^2 pdv = -\int_1^2 \frac{c}{v^\gamma}\, dv$$

where

$$pv^\gamma = p_1 v_1^\gamma = p_2 v_2^\gamma = c$$

$$\therefore \quad u_2 - u_1 = c\frac{v_2^{1-\gamma} - v_1^{1-\gamma}}{\gamma - 1} = \frac{p_2 v_2^\gamma \cdot v_2^{1-\gamma} - p_1 v_1^\gamma \cdot v_1^{1-\gamma}}{\gamma - 1}$$

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For a polytropic process,

$$Q - W_x - \Delta\left[\frac{\overline{V}_2}{2} + gz\right] = \frac{\gamma p_1 v_1}{\gamma - 1}\left[\left(\frac{p_2}{p_1}\right)^{n-1/n} - 1\right]$$

$$= \frac{\gamma p_1 v_1}{\gamma - 1}\left[\left(\frac{v_1}{v_2}\right)^{n-1} - 1\right] \qquad (10.54)$$

Equations (10.52) and (10.54) can be used to determine heat and work quantities for a closed and a steady flow system respectively.

(iii) Integral Property Relations in a Polytropic Process In a pv^n = constant process,

$$\int_1^2 p\,dv = \int_1^2 \frac{p_1 v_1^n}{v^n}\,dv = \frac{p_1 v_1}{n-1}\left[1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right]$$

$$= \frac{p_1 v_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{n-1/n}\right] \qquad (10.55)$$

Similarly,

$$-\int_1^2 v\,dp = \frac{np_1 v_1}{n-1}\left[1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right]$$

$$= \frac{np_1 v_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{n-1/n}\right] \qquad (10.56)$$

The integral of Tds is obtained from the property relation

$$Tds = du + pdv$$

$$\therefore \quad \int_1^2 Tds = \int_1^2 du + \int_1^2 pdv$$

$$= u_2 - u_1 + \int_1^2 pdv$$

Substituting from Eqs. (10.50) and (10.53)

$$\int_1^2 Tds = \frac{\gamma - n}{(\gamma - 1)(n-1)}\, p_1 v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{n-1/n}\right]$$

$$= \frac{\gamma - n}{(\gamma - 1)(n-1)}\, p_1 v_1\left[1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right]$$

$$= \frac{\gamma - n}{(\gamma - 1)(n-1)}\, R(T_1 - T_2) \qquad (10.57)$$

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The expression $\frac{n_k \bar{R} T}{V}$ represents the pressure that the kth gas would exert if it occupied the volume V alone at temperature T. This is called the *partial pressure* of the kth gas and is denoted by p_k. Thus

$$p_1 = \frac{n_1 \bar{R} T}{V}, p_2 = \frac{n_2 \bar{R} T}{V}, \ldots, p_c = \frac{n_c \bar{R} T}{V}$$

and

$$p = p_1 + p_2 + \ldots + p_c \tag{10.81}$$

This is known as *Dalton's law of partial pressures* which states that the total pressure of a mixture of ideal gases is equal to the sum of the partial pressures.
Now

$$V = (n_1 + n_2 + \ldots + n_c) \cdot \frac{\bar{R} T}{p}$$

$$= \Sigma n_K \cdot \frac{\bar{R} T}{p}$$

and the partial pressure of the Kth gas is

$$p_k = \frac{n_k \bar{R} T}{V}$$

Substituting the value of V

$$p_K = \frac{n_K \bar{R} T \cdot p}{\Sigma n_K \cdot \bar{R} T} = \frac{n_K \bar{R} T}{\Sigma n_K} \cdot p$$

Now

$$\Sigma n_K = n_1 + n_2 + \ldots + n_c$$

= Total number of moles of gas

The ratio $\frac{n_k}{\Sigma n_k}$ is called the *mole fraction of the* kth gas, and is denoted by x_K.
Thus

$$x_1 = \frac{n_1}{\Sigma n_K}, x_2 = \frac{n_2}{\Sigma n_K}, \ldots, x_c = \frac{n_c}{\Sigma n_K}$$

and $$p_1 = x_1 p, p_2 = x_2 p, \ldots, p_c = x_c p$$

or $$p_K = x_K \cdot p \tag{10.82}$$

Also

$$x_1 + x_2 + \ldots + x_c = 1 \tag{10.83}$$

In a mixture of gases, if all but one mole fraction is determined, the last can be calculated from the above equation. Again, in terms of masses

$$p_1 V = m_1 R_1 T$$

$$p_2 V = m_2 R_2 T$$

.....

$$p_c V = m_c R_c T$$

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10.11 GIBBS FUNCTION OF A MIXTURE OF INERT IDEAL GASES

From the equations

$$d\bar{h} = \bar{c}_p dT$$

$$d\bar{s} = \bar{c}_p \frac{dT}{T} - \bar{R}\frac{dp}{p}$$

the enthalpy and entropy of 1 mole of an ideal gas at temperature T and pressure p are

$$\bar{h} = \bar{h}_0 + \int c_p\, dT$$

$$\bar{s} = \int \bar{c}_p \frac{dT}{T} + \bar{s}_0 - \bar{R}\ \ln p$$

Therefore, the molar Gibbs function

$$\bar{g} = \bar{h} - Ts$$

$$= \bar{h}_0 + \int \bar{c}_p dT - T\int \bar{c}_p \frac{dT}{T} - T\bar{s}_0 + \bar{R}T \ln p$$

Now $$\int d(uv) = \int u dv + \int v du = uv$$

Let $$u = \frac{1}{T},\ v = \int c_p\, dT$$

Then $$\frac{1}{T}\int c_p\, dT = \int \frac{1}{T}\, c_p\, dT + \int c_p\, dT \left(-\frac{1}{T^2}\right) dT$$

$$= \int \frac{1}{T}\, c_p\, dT - \int \frac{\int c_p dT}{T^2}\, dT$$

$$\int c_p\, dT - T\int c_p \frac{dT}{T} = -T\int \frac{\int c_p dT}{T^2}\, dT$$

Therefore

$$\bar{g} = \bar{h}_0 - T\int \frac{\int \bar{c}_p dT}{T^2}\, dT - T\bar{s}_0 + \bar{R}T\ \ln p$$

$$= \bar{R}T\left(\frac{\bar{h}_0}{\bar{R}T} - \frac{1}{\bar{R}}\int \frac{\int \bar{c}_p dT}{T^2} dT - \frac{\bar{s}_0}{\bar{R}} + \ln p\right)$$

Let

$$\phi = \frac{\bar{h}_0}{\bar{R}T} - \frac{1}{\bar{R}}\int \frac{\int \bar{c}_p dT}{T^2} dT - \frac{\bar{s}_0}{\bar{R}} \qquad (10.100)$$

Thus $$\bar{g} = \bar{R}T\ (\phi + \ln p) \qquad (10.101)$$

where ϕ is a function of temperature only.

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For an isentropic process

$$b \ln T + kT + (a - b) \ln v = \text{constant}$$

$$\therefore \quad T^b \cdot v^{a-b} e^{kT} = \text{constant} \qquad \text{(Q.E.D.)}$$

(b) $R = a - b = 0.946 - 0.662 = 0.284$ kJ/kg K

$$T_2 = 240 + 273 = 513 \text{ K}$$

$$T_1 = \frac{p_1 V_1}{mR} = \frac{5.6 \times 10^3 \times 0.06}{1.5 \times 0.284} = 788.73 \text{ K} = 789 \text{ K}$$

$$TdS = dU + d\,W = 0$$

$$\therefore \quad W_{1-2} = -\int_{T_1}^{T_2} mc_v\, dT$$

$$= 1.5 \int_{513}^{789} (0.662 + 0.0001T)\, dT$$

$$= 1.5\,[0.662\,(789 - 513) + 10^{-4} \times 0.5\,\{(789)^2 - (513)^2\}]$$

$$= 1.5\,(182.71 + 19.97) = \mathbf{304\ kJ}$$

Example 10.4 Show that for an ideal gas, the slope of the constant volume line on the T–s diagram is more than that of the constant pressure line.

Solution We have, for 1 kg of ideal gas

$$Tds = du + pdv$$

$$= c_v\, dT + pdv$$

$$\therefore \quad \left(\frac{\partial T}{\partial s}\right)_v = \frac{T}{c_v}$$

Also

$$Tds = dh - vdp$$

$$= c_p\, dT - vdp$$

$$\therefore \quad \left(\frac{\partial T}{\partial s}\right)_p = \frac{T}{c_p}$$

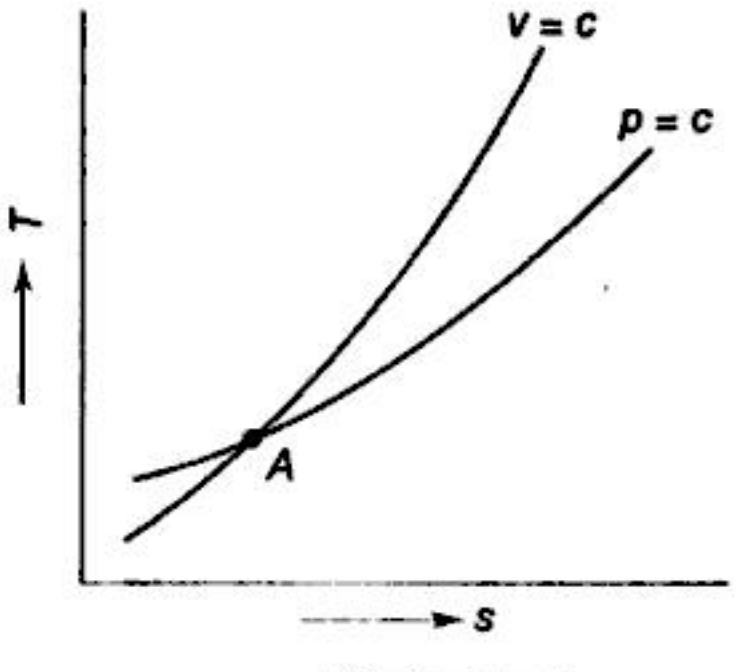

Fig. 10.9

Since $c_p > c_v$, $\frac{T}{c_v} > \frac{T}{c_p}$

$$\therefore \quad \left(\frac{\partial T}{\partial s}\right)_v > \left(\frac{\partial T}{\partial s}\right)_p$$

This is shown in Fig. 10.9. The slope of the constant volume line passing through point A is steeper than that of the constant pressure line passing through the same point. (Q.E.D.)

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(c) The efficiency of the cycle

$$\eta_{\text{cycle}} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{84.12}{137.13} = 1 - 0.61 = \mathbf{0.39, \text{ or } 39\%}$$

Example 10.7 A mass of 0.25 kg of an ideal gas has a pressure of 300 kPa, a temperature of 80°C, and a volume of 0.07 m^3. The gas undergoes an irreversible adiabatic process to a final pressure of 300 kPa and final volume of 0.10 m^3, during which the work done on the gas is 25 kJ. Evaluate the c_p and c_v of the gas and the increase in entropy of the gas.

Solution From

$$p_1V_1 = mRT_1$$

$$R = \frac{300 \times 0.07}{0.25 \times (273 + 80)} = 0.238 \text{ kJ/kg K}$$

Final temperature

$$T_2 = \frac{p_2V_2}{mR} = \frac{300 \times 0.1}{0.25 \times 0.238} = 505 \text{ K}$$

Now

$$Q = (U_2 - U_1) + W = mc_v(T_2 - T_1) + W$$

$$0 = 0.25\ c_v(505 - 353) - 25$$

$$\therefore \quad c_v = \frac{25}{0.25 \times 152} = 0.658 \text{ kJ/kg K}$$

Now $c_p - c_v = R$

$$c_p = 0.658 + 0.238 = 0.896 \text{ kJ/kg K}$$

Entropy change

$$S_2 - S_1 = mc_v \ln \frac{p_2}{p_1} + mc_p \ln \frac{v_2}{v_1}$$

$$= mc_p \ln \frac{V_2}{V_1} = 0.25 \times 0.896 \ln \frac{0.10}{0.07}$$

$$= 0.224 \times 0.3569 = \mathbf{0.08 \text{ kJ/kg K}}$$

Example 10.8 Atmospheric air is sucked in an internal combustion engine cylinder. The pressure, temperature and volume of air after suction are 1 bar, 75°C and 800 cm^3 respectively. The air is then reversibly compressed to 15 bar and 1/8th of its volume. Heat of combustion of fuel is then added to air at constant volume until its pressure is 50 bar. (a) Determine the index of compression process. (b) Calculate the change of entropy of air during each process. (c) Determine if there is any heat exchange between air and cylinder walls during compression. State the direction of heat flow.

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If the mixture is heated at constant pressure, ΔU and ΔH will remain the same. The change in entropy will be

$$S_2 - S_1 = mc_p \ln \frac{T_2}{T_1} - mR \ln \frac{p_2}{p_1}$$

$$= mc_p \ln \frac{T_2}{T_1} = 8 \times 0.92 \ln \frac{313}{293} = \mathbf{0.49\ kJ/kg\ K}$$

Example 10.10 Find the increase in entropy when 2 kg of oxygen at 60°C are mixed with 6 kg of nitrogen at the same temperature. The initial pressure of each constituent is 103 kPa and is the same as that of the mixture.

$$x_{O_2} = \frac{p_{O_2}}{p} = \frac{\frac{2}{32}}{\frac{2}{32} + \frac{6}{28}} = 0.225$$

$$x_{N_2} = \frac{p_{N_2}}{p} = 0.775$$

Entropy increase due to diffusion

$$\Delta S = -m_{O2} R_{O_2} \ln \frac{p_{O_2}}{p} - m_{N_2} R_{N_2} \ln \frac{p_{N_2}}{p}$$

$$= -2\left(\frac{8.3143}{32}\right) \ln 0.225 - 6\left(\frac{8.3143}{28}\right) \ln 0.775 = \mathbf{1.2314\ kJ/kg\ K}$$

Example 10.11 The gas neon has a molecular weight of 20.183 and its critical temperature, pressure and volume are 44.5 K, 2.73 MPa and 0.0416 m^3/kg mol. Reading from a compressibility chart for a reduced pressure of 2 and a reduced temperature of 1.3, the compressibility factor Z is 0.7. What are the corresponding specific volume, pressure, temperature, and reduced volume?

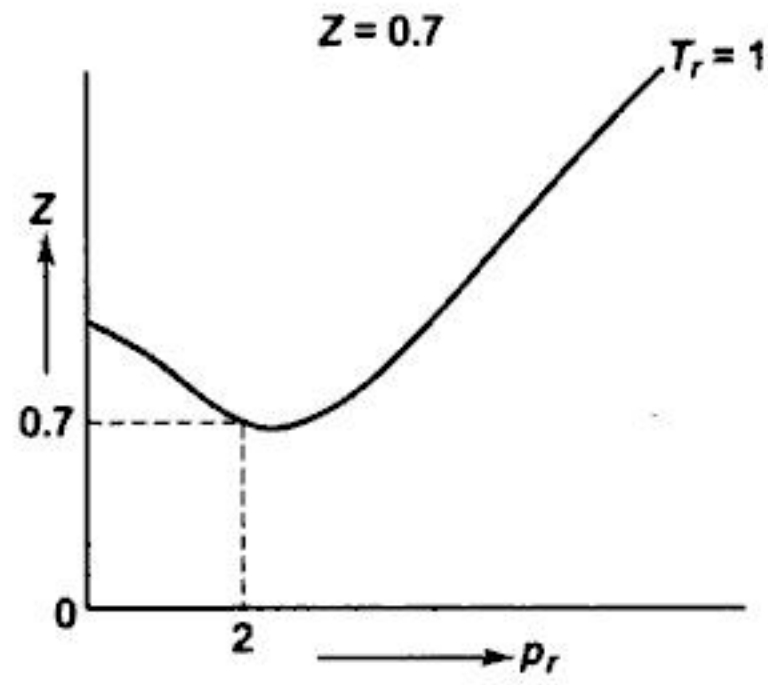

Fig. 10.13

Solution At $p_r = 2$ and $T_r = 1.3$ from chart (Fig. 10.13)

$$Z = 0.7$$

$$p = 2 \times 2.73 = \mathbf{5.46\ MPa}$$

$$\frac{T}{T_c} = 1.3$$

$$T = 1.3 \times 44.5 = \mathbf{57.85\ K}$$

$$p_v = ZRT$$

$$\therefore \quad v = \frac{0.7 \times 8.3143 \times 57.85}{20.183 \times 5.46 \times 10^3} = 3.05 \times 10^{-3}\ m^3/kg$$

$$\therefore \quad v_r = \frac{v}{v_c} = \frac{3.05 \times 10^{-3} \times 20.183}{4.16 \times 10^{-2}} = \mathbf{1.48}$$

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10.20 What are the expressions of work transfer for an ideal gas in a polytropic process, if the gas is: (i) a closed system, and (ii) a steady system?

10.21 Derive the expression of heat transfer for an ideal gas in a polytropic process. What is the polytropic specific heat? What would be the direction of heat transfer if (a) $n > \gamma$, and (b) $n < \gamma$?

10.22 Why is the external work supplied to a compressor equal to $-\int_{p1}^{p2} v\,dp$?

10.23 Why does isothermal compression need minimum work and adiabatic compression maximum work?

10.24 What is the isothermal efficiency of a compressor?

10.25 Write down the van der Waals equation of state. How does it differ from the ideal gas equation of state. What is force of cohesion? What is co-volume?

10.26 What are the two-constant equations of state?

10.27 Give the virial expansions for pv in terms of p and v.

10.28 What are virial coefficients? When do they become zero?

10.29 What is the compressibility factor?

10.30 What are reduced properties?

10.31 What is the generalized compressibility chart?

10.32 What is the law of corresponding states?

10.33 Express the van der Waals constants in terms of critical properties.

10.34 Draw the diagram representing the law of corresponding states in reduced coordinates indicating the isotherms and the liquid and vapour phases.

10.35 Define Boyle temperature? How is it computed?

10.36 State Dalton's law of partial pressures.

10.37 How is the partial pressure in a gas mixture related to the mole fraction?

10.38 How are the characteristic gas constant and the molecular weight of a gas mixture computed?

10.39 What is Gibb's theorem?

10.40 Show that in a diffusion process a gas undergoes a free expansion from the total pressure to the relevant partial pressure.

10.41 Show that in a diffusion process at constant temperature the entropy increases and the Gibbs function decreases.

Problems

10.1 What is the mass of air contained in a room 6 m × 9 m × 4 m if the pressure is 101.325 kPa and the temperature is 25°C? Ans. 256 kg

10.2 The usual cooking gas (mostly methane) cylinder is about 25 cm in diameter and 80 cm in height. It is changed to 12 MPa at room temperature (27°C). (a) Assuming the ideal gas law, find the mass of gas filled in the cylinder. (b) Explain how the actual cylinder contains nearly 15 kg of gas. (c) If the cylinder is to be protected against excessive pressure by means of a fusible plug, at what temperature should the plug melt to limit the maximum pressure to 15 MPa?

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10.30 A certain mass of sulphur dioxide (SO_2) is contained in a vessel of 0.142 m^3 capacity, at a pressure and temperature of 23.1 bar and 18°C respectively. A valve is opened momentarily and the pressure falls immediately to 6.9 bar. Sometimes later the temperature is again 18°C and the pressure is observed to be 9.1 bar. Estimate the value of specific heat ratio.

Ans. 1.29

10.31 A gaseous mixture contains 21% by volume of nitrogen, 50% by volume of hydrogen, and 29% by volume of carbon-dioxide. Calculate the molecular weight of the mixture, the characteristic gas constant R for the mixture and the value of the reversible adiabatic index γ. (At 10°C, the c_p values of nitrogen, hydrogen, and carbon dioxide are 1.039, 14.235, and 0.828 kJ/kg K respectively.)

A cylinder contains 0.085 m^3 of the mixture at 1 bar and 10°C. The gas undergoes a reversible non-flow process during which its volume is reduced to one-fifth of its original value. If the law of compression is $pv^{1.2}$ = constant, determine the work and heat transfer in magnitude and sense and the change in entropy.

Ans. 19.64 kg/kg mol, 0.423 kJ/kg K, 1.365,
– 16 kJ, – 7.24 kJ, – 0.31 kJ/kg K

10.32 Two moles of an ideal gas at temperature T and pressure p are contained in a compartment. In an adjacent compartment is one mole of an ideal gas at temperature $2T$ and pressure p. The gases mix adiabatically but do not react chemically when a partition separating the compartments is withdrawn. Show that the entropy increase due to the mixing process is given by

$$R\left(\ln\frac{27}{4}+\frac{\gamma}{\gamma-1}\ln\frac{32}{27}\right)$$

provided that the gases are different and that the ratio of specific heat γ is the same for both gases and remains constant.

What would the entropy change be if the mixing gases were of the same species?

10.33 n_1 moles of an ideal gas at pressure p_1 and temperature T are in one compartment of an insulated container. In an adjoining compartment, separated by a partition, are n_2 moles of an ideal gas at pressure p_2 and temperature T. When the partition is removed, calculate (a) the final pressure of the mixture, (b) the entropy change when the gases are identical, and (c) the entropy change when the gases are different. Prove that the entropy change in (c) is the same as that produced by two independent free expansions.

10.34 Assume that 20 kg of steam are required at a pressure of 600 bar and a temperature of 750°C in order to conduct a particular experiment. A 140-litre heavy duty tank is available for storage.

Predict if this is an adequate storage capacity using:

(a) the ideal gas theory,
(b) the compressibility factor chart,
(c) the van der Waals equation with $a = 5.454$ (litre)^2 atm/(g mol)^2, $b = 0.03042$ litres/gmol for steam,
(d) the Mollier chart
(e) the steam tables.

Estimate the error in each.

10.35 Estimate the pressure of 5 kg of CO_2 gas which occupies a volume of 0.70 m^3 at 75°C, using the Beattie-Bridgeman equation of state.

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By substituting in the energy Eq. (11.13)

$$u = \frac{T}{3}\frac{du}{dT} - \frac{u}{3}$$

$$\therefore \qquad \frac{du}{u} = 4\,\frac{dT}{T}$$

or $$\ln u = \ln T^4 + \ln b$$

or $$u = bT^4$$

where b is a constant. This is known as the *Stefan-Boltzmann Law.*

Since $$U = uV = VbT^4$$

$$\left(\frac{\partial U}{\partial T}\right)_V = C_v = 4VbT^3$$

and $$\left(\frac{\partial p}{\partial T}\right)_V = \frac{1}{3}\frac{du}{dT} = \frac{4}{3}\,bT^3$$

from the first TdS equation

$$TdS = C_v\,dT + T\left(\frac{\partial p}{\partial T}\right)_V dV$$

$$= 4VbT^3\,dT + \frac{4}{3}\,bT^4 \cdot dV$$

For a reversible isothermal change of volume, the heat to be supplied reversibly to keep temperature constant

$$Q = \frac{4}{3}\,bT^4\,\Delta V$$

For a reversible adiabatic change of volume

$$\frac{4}{3}\,bT^4\,dV = -4VbT^3\,dT$$

$$\therefore \qquad \frac{dV}{V} = -3\,\frac{dT}{T}$$

or $$VT^3 = \text{const.}$$

If the temperature is one-half the original temperature, the volume of black-body radiation is to be increased adiabatically eight times its original volume so that the radiation remains in equilibrium with matter at that temperature.

11.7 JOULE-KELVIN EFFECT

A gas is made to undergo continuous throttling process by a valve, as shown in Fig. 11.2. The pressures and temperatures of the gas in the insulated pipe upstream and downstream of the valve are measured with suitable manometers and thermometers.

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So the Joule-Kelvin coefficient μ_J is given by, from Equation (11.17)

$$\mu_J = \frac{1}{c_p}\left[T\left(\frac{\partial v}{\partial T}\right)_p - v\right]$$

or $$\mu_J = \frac{v}{c_p}\left[\beta T - 1\right]$$

For an ideal gas, $$\beta = \frac{1}{T} \text{ and } \mu_J = 0$$

There are two inversion temperatures for each pressure, e.g. T_1 and T_2 at pressure p (Fig. 11.4).

11.8 CLAUSIUS-CLAPEYRON EQUATION

During phase transitions like melting, vaporization and sublimation, the temperature and pressure remain constant, while the entropy and volume change. If x is the fraction of initial phase i which has been transformed into final phase f, then

$$s = (1 - x)\, s^{(i)} + x s^{(f)}$$

$$v = (1 - x)\, v^{(i)} + x v^{(f)}$$

where s and v are linear function of x.

For reversible phase transition, the heat transferred per mole (or per kg) is the latent heat, given by

$$l = T\{s^{(f)} - s^{(i)}\}$$

which indicates the change in entropy.

Now $$dg = -sdT + vdp$$

or $$s = -\left(\frac{\partial g}{\partial T}\right)_p$$

and $$v = \left(\frac{\partial g}{\partial p}\right)_T$$

A *phase change of the first order* is known as any phase change that satisfies the following requirements:

(a) There are changes of entropy and volume.

(b) The first-order derivatives of Gibbs function change discontinuously.

Let us consider the first-order phase transition of one mole of a substance from phase i to phase f. Using the first TdS equation

$$Tds = c_v\, dT + T\left(\frac{\partial p}{\partial T}\right)_v dv$$

for the phase transition which is reversible, isothermal and isobaric, and integrating over the whole change of phase, and since $\left(\frac{\partial p}{\partial T}\right)_v$ is independent of v

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Path 0-*a*-*B*:
From Eq. 11.25,

$$h_a - h_0 = \int_{T_0}^{T} c_p \, dT$$

$$h - h_a = \int_{p_0}^{p} \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right]_T dp$$

On addition

$$h - h_0 = \left[\int_{T_0}^{T} c_p dT\right]_{p_0} + \left\{\int_{p_0}^{p} \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right] dp \right\}_T \qquad (11.27)$$

Similarly, for
Path 0-*b*-*B*:

$$h - h_0 = \left\{\int_{p_0}^{p} \left[v - T\left(\frac{\partial v}{\partial T}\right)_p \right] dp \right\}_{T_0} + \left[\int_{T_0}^{T} c_p \, dT\right]_p \qquad (11.28)$$

Equation (11.27) is preferred to Eq. (11.28) since c_p at lower pressure can be conveniently measured. Now,

$$\int_{p_0 v_a}^{pv} d\,(pv) = \int_{v_a}^{v} p \, dv + \int_{p_0}^{p} v \, dp$$

$$\int_{v_a}^{v} v \, dp = pv - p_0 v_a - \left[\int_{v_a}^{v} p dv\right]_T \qquad (11.29)$$

Again,

$$\left[\frac{\partial v}{\partial T}\right]_p \left[\frac{\partial T}{\partial p}\right]_v \left[\frac{\partial p}{\partial v}\right]_T = -1$$

$$\therefore \qquad \left[\frac{\partial v}{\partial T}\right]_p = -\left[\frac{\partial p}{\partial T}\right]_v \left[\frac{\partial v}{\partial p}\right]_T$$

Substituting in Eq. (11.27),

$$h - h_0 = \left[\int_{T_0}^{T} c_p dT\right]_{p_0} + pv - p_0 v_a - \left[\int_{v_a}^{v} p dv\right]_T$$

$$+ \left\{\int_{p_0}^{p} T\left(\frac{\partial p}{\partial T}\right)_p \left(\frac{\partial v}{\partial p}\right)_T dp\right\}_T$$

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$$\therefore \qquad \left(\frac{\partial U}{\partial S}\right)_{V,n_i} = T, \quad \text{and} \quad \left(\frac{\partial U}{\partial V}\right)_{S,n_i} = -p$$

$$\therefore \qquad dU = TdS - pdV + \sum_{i=1}^{K}\left(\frac{\partial U}{\partial n_i}\right)_{S,V,n_j} dn_i \tag{11.32}$$

Molal *chemical potential*, μ, of component i is defined as

$$\mu_i = \left(\frac{\partial U}{\partial n_i}\right)_{S,V,n_j}$$

signifying the change in internal energy per unit mole of component i when S, V, and the number of moles of all other components are constant.

$$\therefore \qquad dU = TdS - pdV + \sum_{i=1}^{K} \mu_i\, dn_i$$

or
$$TdS = dU + pdV - \sum_{i=1}^{K} \mu_i\, dn_i \tag{11.33}$$

This is known as *Gibbs entropy equation.*

In a similar manner

$$G = G\,(p, T, n_1, n_2, \ldots, n_K)$$

or
$$dG = \left(\frac{\partial G}{\partial p}\right)_{T,n_i} dp + \left(\frac{\partial G}{\partial T}\right)_{p,n_i} dT + \sum_{i=1}^{K}\left(\frac{\partial G}{\partial n_i}\right)_{T,p,n_j} dn_i$$

$$= Vdp - SdT + \sum_{i=1}^{K}\left(\frac{\partial G}{\partial n_i}\right)_{T,p,n_j} dn_i \tag{11.34}$$

Since
$$G = U + pV - TS$$

$$d(U + pV - TS) = VdP - SdT + \sum_{i=1}^{K}\left(\frac{\partial G}{\partial n_i}\right)_{T;p,n_j} dn_i$$

or
$$dU + pdV + Vdp - TdS - SdT$$

$$= Vdp - SdT + \sum_{i=1}^{K}\left(\frac{\partial G}{\partial n_i}\right)_{T,p,n_j} dn_i$$

or
$$dU = TdS - pdV + \sum_{i=1}^{K}\left(\frac{\partial G}{\partial n_i}\right)_{T,p,n_j} dn_i$$

Comparing this equation with Eq. (11.32)

$$\left(\frac{\partial U}{\partial n_i}\right)_{S,V,n_j} = \left(\frac{\partial G}{\partial n_i}\right)_{T,p,n_j} = \mu_i$$

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component in phase a must be equal to the chemical potential of the same component in all other phases.

11.13 GIBBS PHASE RULE

Let us consider a heterogeneous system of C chemical constituents which do not combine chemically with one another. Let us suppose that there are ϕ phases, and every constituent is present in each phase. The constituents are denoted by subscripts and the phases by superscripts. The Gibbs function of the whole heterogeneous system is

$$G_{T,p} = \sum_{i=1}^{C} n_i^{(1)} \mu_i^{(1)} + \sum_{i=1}^{C} n_i^{(2)} \mu_i^{(2)} + \ldots + \sum_{i=1}^{C} n_i^{(\phi)} \mu_i^{(\phi)}$$

G is a function of T, p, and the n's of which there are $C\phi$ in number. Since there are no chemical reactions, the only way in which the n's may change is by the transport of the constituents from one phase to another. In this case the total number of moles of each constituent will remain constant.

$$n_1^{(1)} + n_1^{(2)} + \ldots + n_1^{(\phi)} = \text{constant}$$
$$n_2^{(1)} + n_2^{(2)} + \ldots + n_2^{(\phi)} = \text{constant}$$
$$\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$
$$n_C^{(1)} + n_C^{(2)} + \ldots + n_C^{(\phi)} = \text{constant}$$

These are the *equations of constraint.*

At chemical equilibrium, G will be rendered a minimum at constant T and p, subject to these equations of constraint. At equilibrium, from Equation (11.43).

$$\mu_{ij} = \mu_{ia}$$
$$\mu_1^{(1)} = \mu_1^{(2)} = \ldots = \mu_1^{(\phi)}$$
$$\mu_2^{(1)} = \mu_2^{(2)} = \ldots = \mu_2^{(\phi)} \qquad (11.44)$$
$$\ldots\ldots\ldots\ldots\ldots\ldots$$
$$\mu_C^{(1)} = \mu_C^{(2)} = \ldots = \mu_C^{(\phi)}$$

These are known as the *equations of phase equilibrium.* The equations of the phase equilibrium of one constituent are $(\phi - 1)$ in number. Therefore, for C constituents, there are $C(\phi - 1)$ such equations.

When equilibrium has been reached, there is no transport of matter from one phase to another. Therefore, in each phase, $\Sigma x = 1$. For ϕ phases, there are ϕ such equations available.

The state of the system at equilibrium is determined by the temperature, pressure, and $C\phi$ mole fractions. Therefore

Total number of variables = $C\phi + 2$.

Among these variables, there are $C(\phi - 1)$ equations of phase equilibrium and ϕ equations of $\Sigma x = 1$ type. Therefore

Total number of equations = $C(\phi - 1) + \phi$

If the number of variables is equal to the number of equations, the system is *nonvariant*. If the number of variables exceeds the number of equations by one, then the system is called *monovariant* and is said to have a variance of 1.

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The values of U_1, U_2, V_1, and V_2 would change in such a way as to maximize the value of entropy. Therefore, when the equilibrium condition is achieved

$$dS = 0$$

for the whole system. Since

$$S = S_1 + S_2$$
$$= S_1(U_1, V_1, ..., N_{k1} ...) + S_2 (U_2, V_2, ..., N_{k2} ...)$$

$$\therefore \quad dS = \left(\frac{\partial S_1}{\partial U_1}\right)_{V_1 ..., N_{k1} ...} dU_1 + \left(\frac{\partial S_1}{\partial V_1}\right)_{U_1 ..., N_{k1} ...} dV_1$$

$$+ \left(\frac{\partial S_2}{\partial U_2}\right)_{V_2 ..., N_{k2} ...} dU_2 + \left(\frac{\partial S_2}{\partial V_2}\right)_{V_2 ..., N_{k2} ...} dV_2$$

$$= \frac{1}{T_1} dU_1 + \frac{p_1}{T_1} dV_1 + \frac{1}{T_2} dU_2 + \frac{p_2}{T_2} dV_2$$

Since $dU_1 + dU_2 = 0$ and $dV_1 + dV_2 = 0$

$$\therefore \quad dS = \left(\frac{1}{T_1} - \frac{1}{T_2}\right) dU_1 + \left(\frac{p_1}{T_1} - \frac{p_2}{T_2}\right) dV_1 = 0$$

Since the expression must vanish for arbitrary and independent values of dU_1 and dV_1

$$\frac{1}{T_1} - \frac{1}{T_2} = 0 \text{ and } \frac{p_1}{T_1} - \frac{p_2}{T_2} = 0$$

or $$p_1 = p_2 \text{ and } T_1 = T_2$$

$\therefore$ These are the conditions of mechanical and thermal equilibrium.

(b) We will consider the equilibrium state of two simple subsystems (Fig. 11.16 (b)) connected by a rigid and diathermal wall, permeable to one type of material (N_1) and impermeable to all others (N_2, N_3, ... N_r). We thus seek the equilibrium values of U_1 and U_2, and of N_{1-1} and N_{1-2} (i.e. material N_1 in subsystems 1 and 2 respectively.)

At equilibrium, an infinitesimal change in entropy is zero

$$dS = 0$$

Now $$dS = dS_1 + dS_2$$

$$= \left(\frac{\partial S_1}{\partial U_1}\right)_{V_1, N_{1-1}, ...} dU_1 + \left(\frac{\partial S_1}{\partial N_{1-1}}\right)_{U_1, V_1, N_{1-2}, ...} dN_{1-1}$$

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11.18 Show that the change in latent heat L with temperature is given by the following relation

$$\left(\frac{dL}{dT}\right) = (C_p''' - C_p'') + \frac{L}{T} - \frac{v'''\beta''' - v''\beta''}{v''' - v''} L$$

11.19 Show that for a van der Waals gas, the Joule–Thomson coefficient is given by

$$\mu_j = \frac{v}{c_p}\left[\frac{2a(v-b)^2 - RTbv^2}{RTv^3 - 2a(v-b)^2}\right]$$

11.20 At 273.15 K the specific volumes of water and ice are 0.001 and 0.001091 m^3/kg and the latent heat of fusion of ice is 334 kJ/kg. Determine the melting point increase due to increase of pressure by 1 atm (101.325 kPa)

Ans. 0.00753 K

11.21 Calculate the latent heat of vaporization of steam formed by boiling water under a pressure of 101.325 kPa. At a pressure near this, a rise of temperature of 1 K causes an increase of vapour pressure of 3.62 kPa. Ans. 2257 kJ/kg

11.22 It is known that radiation exerts a pressure $p = 1/3\ u$, where u is the energy per unit volume.

(a) Show that

$$du = Tds + \frac{1}{V}\left(Ts - \frac{4}{3}u\right) dV$$

where s is the entropy per unit volume.

(b) Assuming u and s as functions of temperature only, show that

(i) $u = As^{4/3}$

(ii) $s = \frac{4}{3} aT^3$

(iii) $u = aT^4$

where A is the constant of integration and $a = 81/256\ A^3$.

(c) Show that the average time radiation remains in a spherical enclosure of radius r is given by

$$t = \frac{4r}{3c}$$

where c is the speed of radiation.

(d) If E_B is the energy emitted per unit area of spherical surface per unit time, show that

$$E_B = \sigma T^4$$

where $\sigma = ac/4$ and T is the temperature of the surface.

11.23 Show that the inversion temperature of a van der Waals gas is given by $T_i = 2a/bR$.

11.24 Show that:

(a) $\left(\frac{\partial u}{\partial v}\right)_T = T^2\left[\frac{\partial (p/T)}{\partial T}\right]_v$

(b) $\left(\frac{\partial h}{\partial p}\right)_T = T^2\left[\frac{\partial (v/T)}{\partial T}\right]_v$

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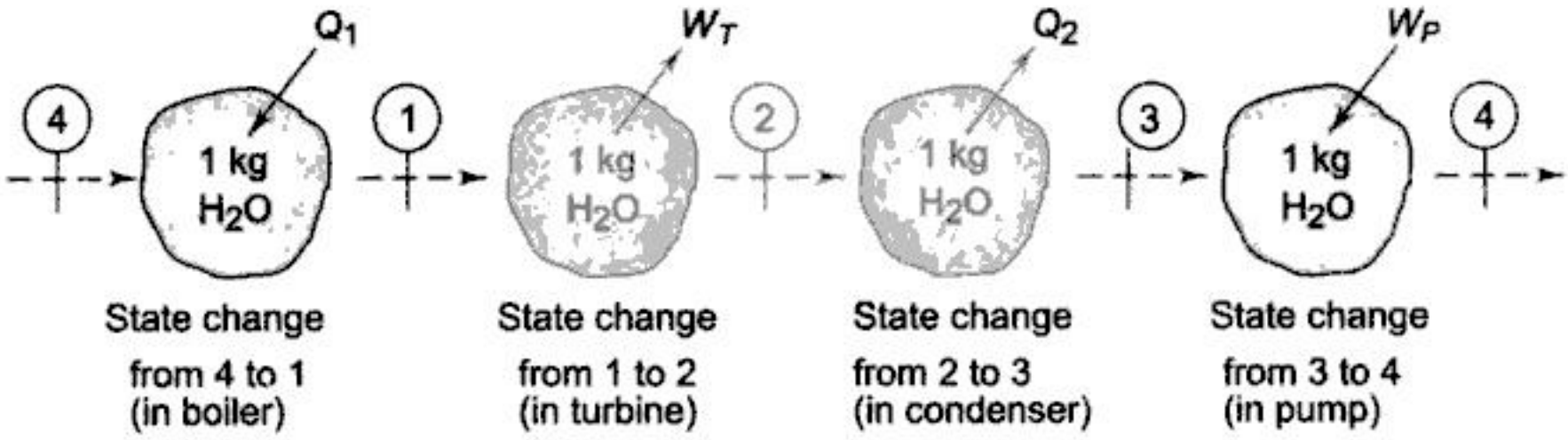

Fig. 12.2 *One kg H_2O Executing a Heat Engine Cycle*

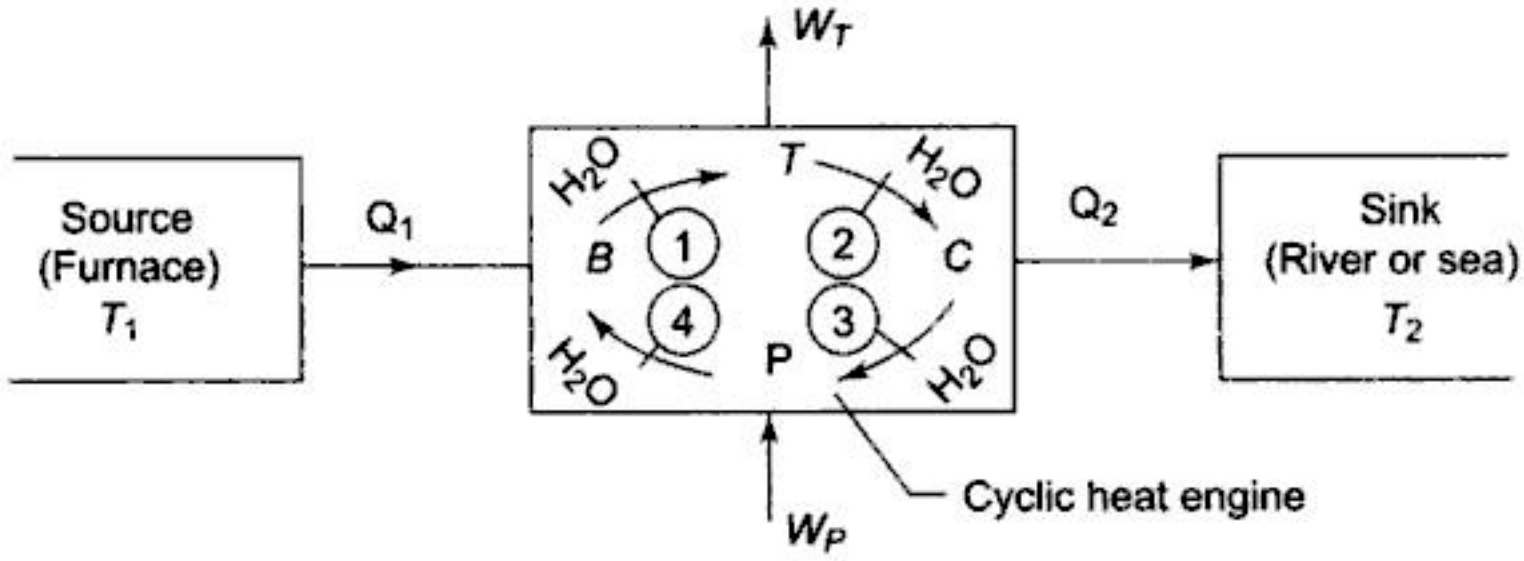

Fig. 12.3 *Cyclic Heat Engine With Water as the Working Fluid*

The efficiency of the vapour power cycle would be given by

$$\eta_{\text{cycle}} = \frac{W_{\text{net}}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$

$$= 1 - \frac{Q_2}{Q_1} \qquad (12.1)$$

12.2 RANKINE CYCLE

For each process in the vapour power cycle, it is possible to assume a hypothetical or ideal process which represents the basic intended operation and involves no extraneous effects. For the steam boiler, this would be a reversible constant pressure heating process of water to form steam, for the turbine the ideal process would be a reversible adiabatic expansion of steam, for the condenser it would be a reversible constant pressure heat rejection as the steam condenses till it becomes saturated liquid, and for the pump, the ideal process would be the reversible adiabatic compression of this liquid ending at the initial pressure. When all these four processes are ideal, the cycle is an ideal cycle, called a *Rankine cycle*. This is a reversible cycle. Figure 12.4 shows the flow diagram of the Rankine cycle, and in Fig. 12.5, the cycle has been plotted on the p-v, T-s, and h-s planes. The numbers on the plots correspond to the numbers on the flow diagram. For any given pressure, the steam approaching the turbine may be dry saturated (state 1) wet (state 1′), or superheated (state 1″), but the fluid approaching the pump is, in each case, saturated

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$$\eta_{th} = \frac{W_{net}}{Q_1}$$

where the work and heat quantities are the measured values for the actual cycle, which are different from the corresponding quantities of the ideal cycle.

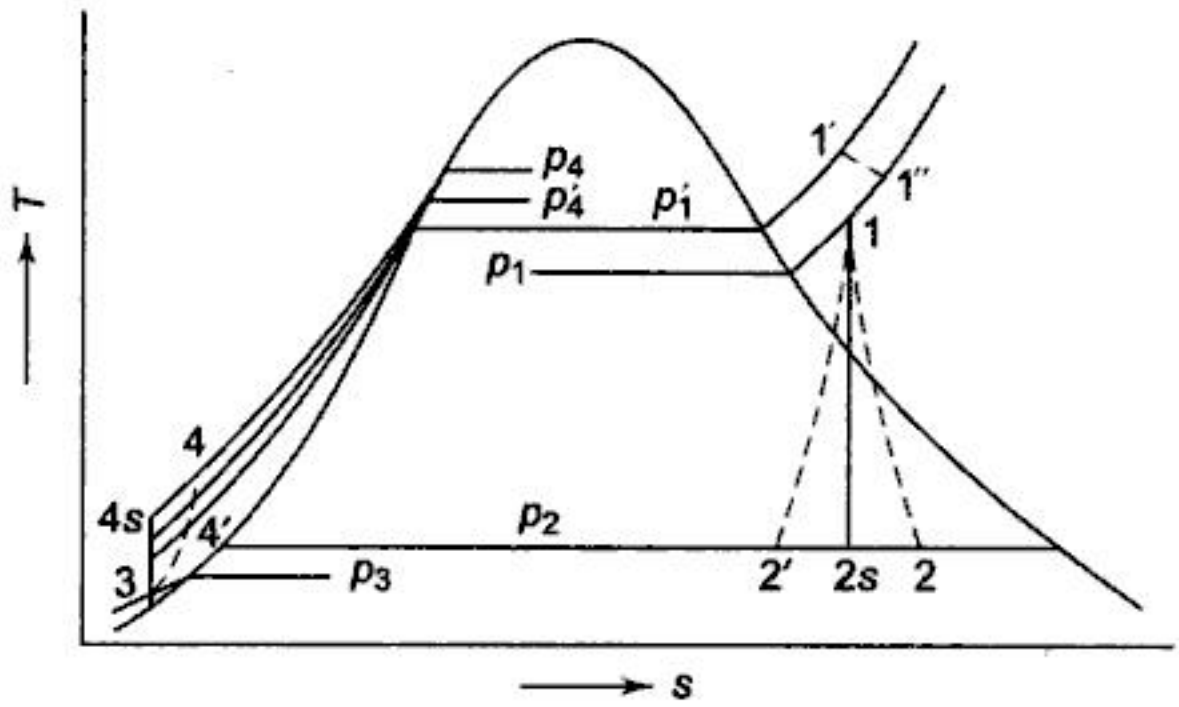

Fig. 12.8 *Various Losses on T-s Plot*

12.3.1 Piping Losses

Pressure drop due to friction and heat loss to the surroundings are the most important piping losses. States 1′ and 1 (Fig. 12.8) represent the states of the steam leaving the boiler and entering the turbine respectively, 1′ – 1″ represents the frictional losses, and 1″ – 1 shows the constant pressure heat loss to the surroundings. Both the pressure drop and heat transfer reduce the availability of steam entering the turbine.

A similar loss is the pressure drop in the boiler and also in the pipeline from the pump to the boiler. Due to this pressure drop, the water entering the boiler must be pumped to a much higher pressure than the desired steam pressure leaving the boiler, and this requires additional pump work.

12.3.2 Turbine Losses

The losses in the turbine are those associated with frictional effects and heat loss to the surroundings. The steady flow energy equation for the turbine in Fig. 12.7 gives

$$h_1 = h_2 + W_T + Q_{loss}$$

$$\therefore \quad W_T = h_1 - h_2 - Q_{loss} \tag{12.10}$$

For the reversible adiabatic expansion, the path will be 1 – 2s. For an ordinary real turbine the heat loss is small, and W_T is $h_1 - h_2$, with Q_2 equal to zero. Since actual turbine work is less than the reversible ideal work output, h_2 is greater than h_{2s}. However, if there is heat loss to the surroundings, h_2 will decrease, accompanied by a decrease in entropy. If the heat loss is large, the end state of steam from the turbine may be 2′. It may so happen that the entropy increase due to frictional

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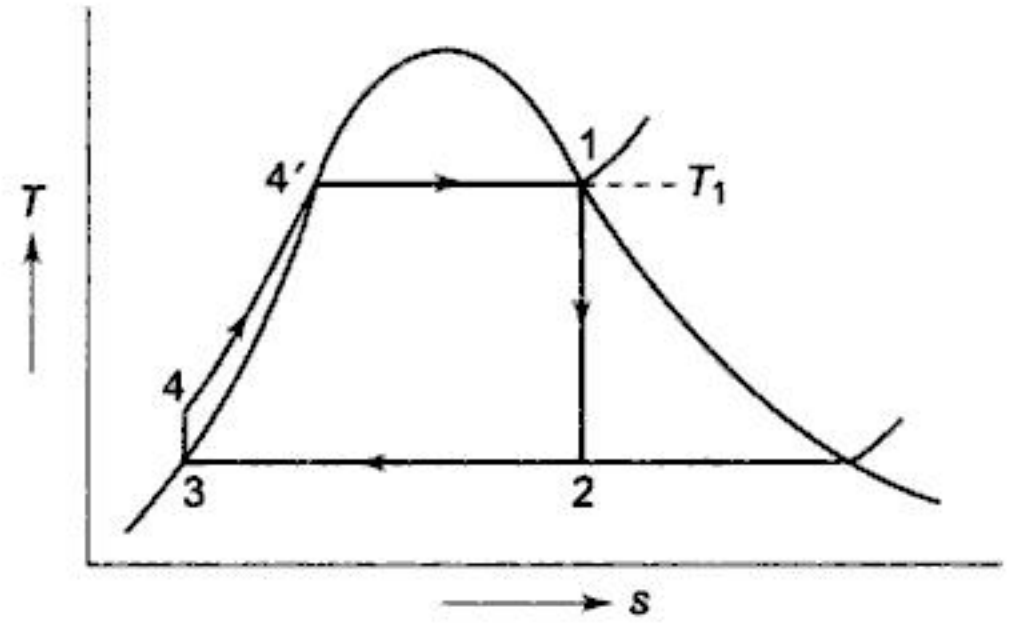

Fig. 12.15 *Simple Rankine Cycle*

The unique feature of the ideal regenerative cycle is that the condensate, after leaving the pump circulates around the turbine casing, counterflow to the direction of vapour flow in the turbine (Fig. 12.16). Thus it is possible to transfer heat from the vapour as it flows through the turbine to the liquid flowing around the turbine. Let us assume that this is a reversible heat transfer, i.e. at each point the temperature of the vapour is only infinitesimally higher than the temperature of the liquid. The process 1–2′ (Fig. 12.17) thus represents reversible expansion of steam in the turbine with reversible heat rejection. For any small step in the process of heating the water,

$$\Delta T \text{ (water)} = -\Delta T \text{ (steam)}$$

and

$$\Delta s \text{ (water)} = -\Delta s \text{ (steam)}$$

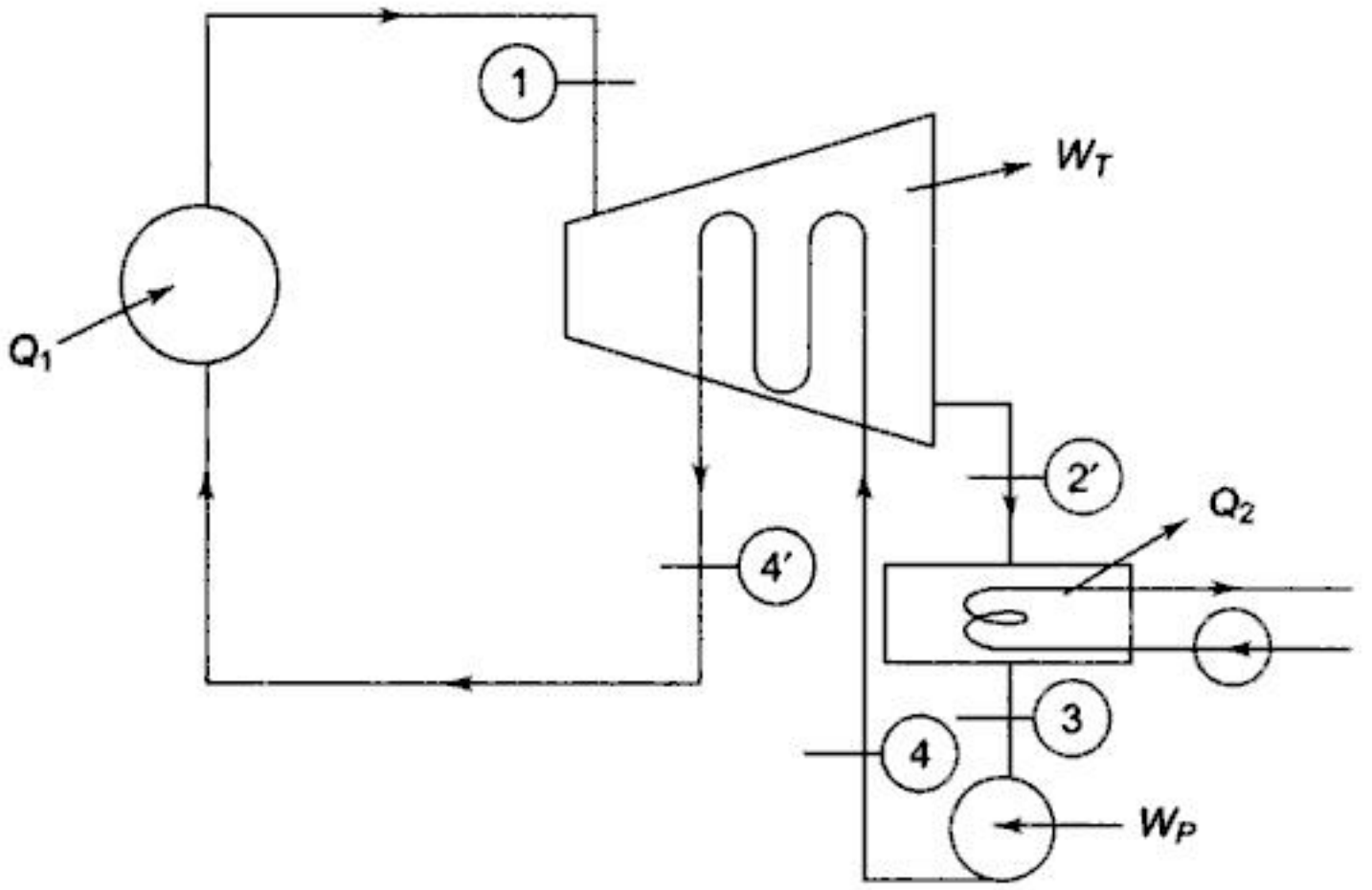

Fig. 12.16 *Ideal Regenerative Cycle—Basic Scheme*

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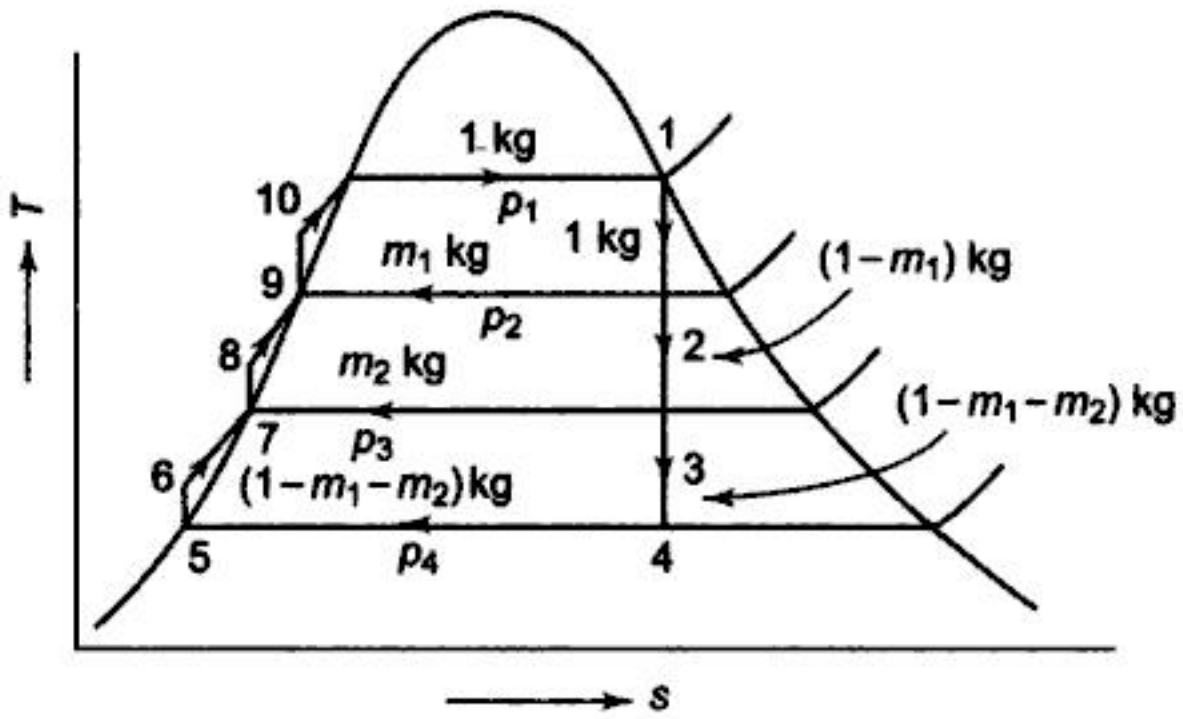

Fig. 12.19 *(a) Regenerative Cycle on T-s Plot with Decreasing Mass of Fluid*
(b) Regenerative Cycle on T-s Plot for Unit Mass of Fluid

Path 1–2–3–4 in Fig. 12.19 represents the states of a decreasing mass of fluid. For 1 kg of steam, the states would be represented by the path 1–2′–3′–4′. From Eq. (12.18),

$$W_T = (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$

$$= (h_1 - h_2) + (h_{2'} - h_{3'}) + (h_{3'} - h_{4'}) \quad (12.26)$$

where $$(1 - m_1)(h_2 - h_3) = 1\,(h_{2'} - h_{3'}) \quad (12.27)$$

$$(1 - m_1 - m_2)(h_3 - h_4) = 1\,(h_{3''} - h_4) \quad (12.28)$$

The cycle 1–2–2′–3′–3″–4′–5–6–7–8–9–10–1 represents 1 kg of working fluid. The heat released by steam condensing from 2 to 2′ is utilized in heating up the water from 8 to 9.

$$\because \quad 1(h_2 - h_{2'}) = (h_9 - h_8) \quad (12.29)$$

Similarly

$$1(h_{3'} - h_{3''}) = 1(h_7 - h_6) \quad (12.30)$$

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Similarly, the exergy loss rate with the exhaust stream is:

$$a_{f_2} = w_g\, c_{p_g}\, T_0 \left[\frac{T_e}{T_0} - 1 - \ln \frac{T_e}{T_0}\right]$$

Net exergy input rate in the steam generation process:

$$a_{i_1} = a_{f_1} - a_{f_2}$$

The exergy utilization rate in the steam generation is:

$$a_{f_u} = w_s\, [(h_1 - h_4)] - T_0(s_1 - s_4)]$$

Rate of exergy loss in the steam generator:

$$I = a_{f_i} - a_{f_u}$$

The useful mechanical power output:

$$= W_{\text{net}} = w_s[(h_1 - h_2) - (h_4 - h_3)]$$

Exergy flow rate of the wet steam to the condenser:

$$a_{f_c} = w_s\, [(h_2 - h_3) - T_0(s_2 - s_3)]$$

Second law efficiency, $\eta_{\text{II}} = \dfrac{W_{\text{net}}}{a_{f_1} - a_{f_2}}$

Exergy flow or Grassman diagram is shown in Fig. 12.28 (c). The energy disposition diagram (b) shows that the major energy loss (~60%) takes place in the condenser. This energy rejection, however, occurs at a temperature close to the ambient temperature, and, therefore, corresponds to a very low exergy value (~4%). The major exergy destruction due to irreversibilities takes place in the steam generation. To improve the performance of the steam plant the finite source temperatures must be closer to the working fluid temperatures to reduce thermal irreversibility.

12.12 CHARACTERISTICS OF AN IDEAL WORKING FLUID IN VAPOUR POWER CYCLES

There are certain drawbacks with steam as the working substance in a power cycle. The maximum temperature that can be used in steam cycles consistent with the best available material is about 600°C, while the critical temperature of steam is 375°C, which necessitates large superheating and permits the addition of only an infinitesimal amount of heat at the highest temperature.

High moisture content is involved in going to higher steam pressures in order to obtain higher mean temperature of heat addition (T_{m_1}). The use of reheat is thus necessitated. Since reheater tubes are costly, the use of more than two reheats is hardly recommended. Also, as pressure increases, the metal stresses increase, and the thicknesses of the walls of boiler drums, tubes, pipe lines, etc. increase not in proportion to pressure increase, but much faster, because of the prevalence of high temperature.

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The mercury-steam cycle represents the two-fluid cycles. The mercury cycle is called the *topping cycle* and the steam cycle is called the *bottoming cycle*. If a sulphur dioxide cycle is added to it in the low temperature range, so that the heat released during the condensation of steam is utilized in forming sulphur dioxide vapour which expands in another turbine, then the mercury-steam-sulphur dioxide cycle is a three-fluid or tertiary cycle. Similarly, other liquid metals, apart from mercury, like sodium or potassium, may be considered for a working fluid in the topping cycle. Apart from SO_2 other refrigerants (ammonia, freons, etc.) may be considered as working fluids for the bottoming cycle.

Since the possibilities of improving steam cycles are diminishing, and the incentives to reduce fuel cost are very much increasing, coupled cycles, like the mercury-steam cycle, may receive more favourable consideration in the near future.

12.14 THERMODYNAMICS OF COUPLED CYCLES

If two cycles are coupled in series where heat lost by one is absorbed by the other (Fig. 12.32), as in the mercury-steam binary cycle, let η_1 and η_2 be the efficiencies of the topping and bottom cycles respectively, and η be the overall efficiency of the combined cycle.

$$\eta_1 = 1 - \frac{Q_2}{Q_1} \text{ and } \eta_2 = 1 - \frac{Q_3}{Q_2}$$

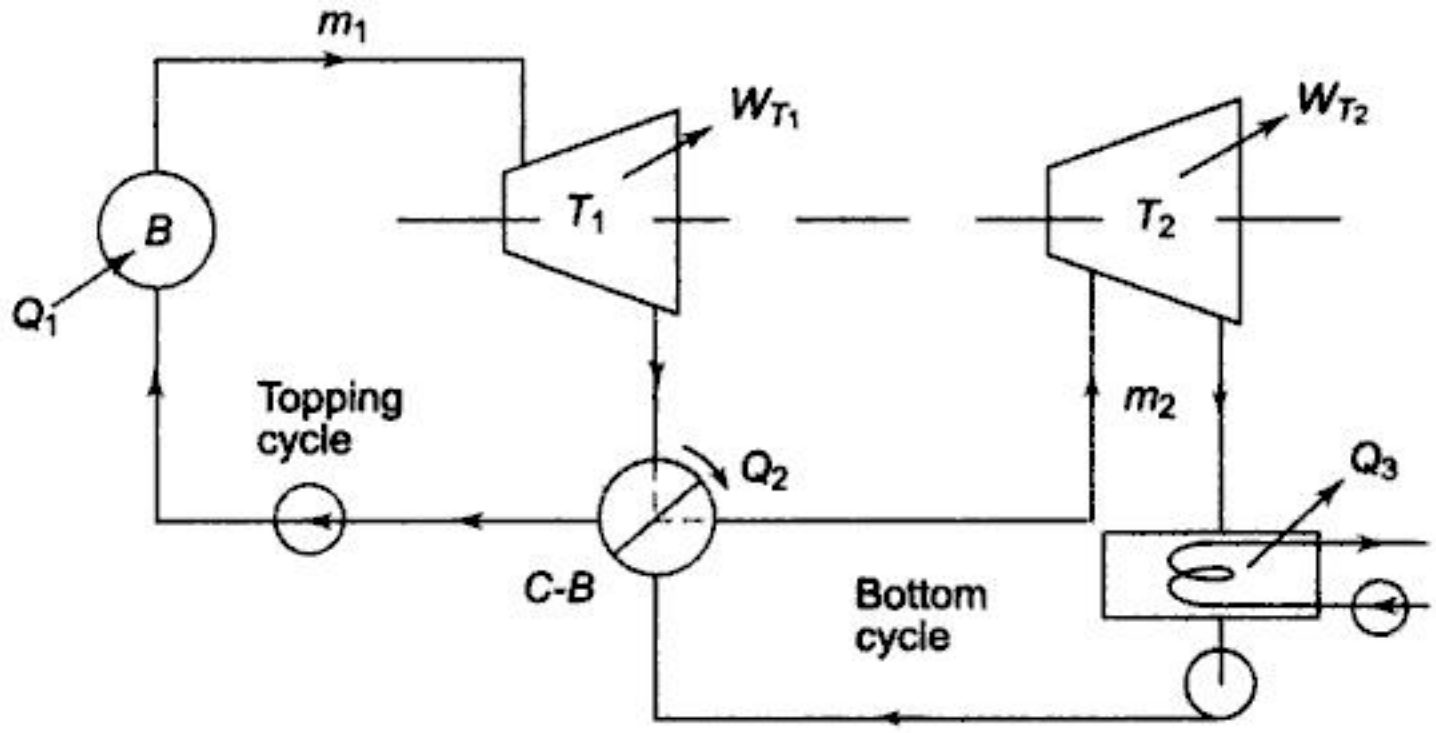

Fig. 12.32 *Two Vapour Cycles Coupled in Series*

or $$Q_2 = Q_1(1 - \eta_1) \text{ and } Q_3 = Q_2(1 - \eta_2)$$

Now $$\eta = 1 - \frac{Q_3}{Q_1} = 1 - \frac{Q_2(1-\eta_2)}{Q_1}$$

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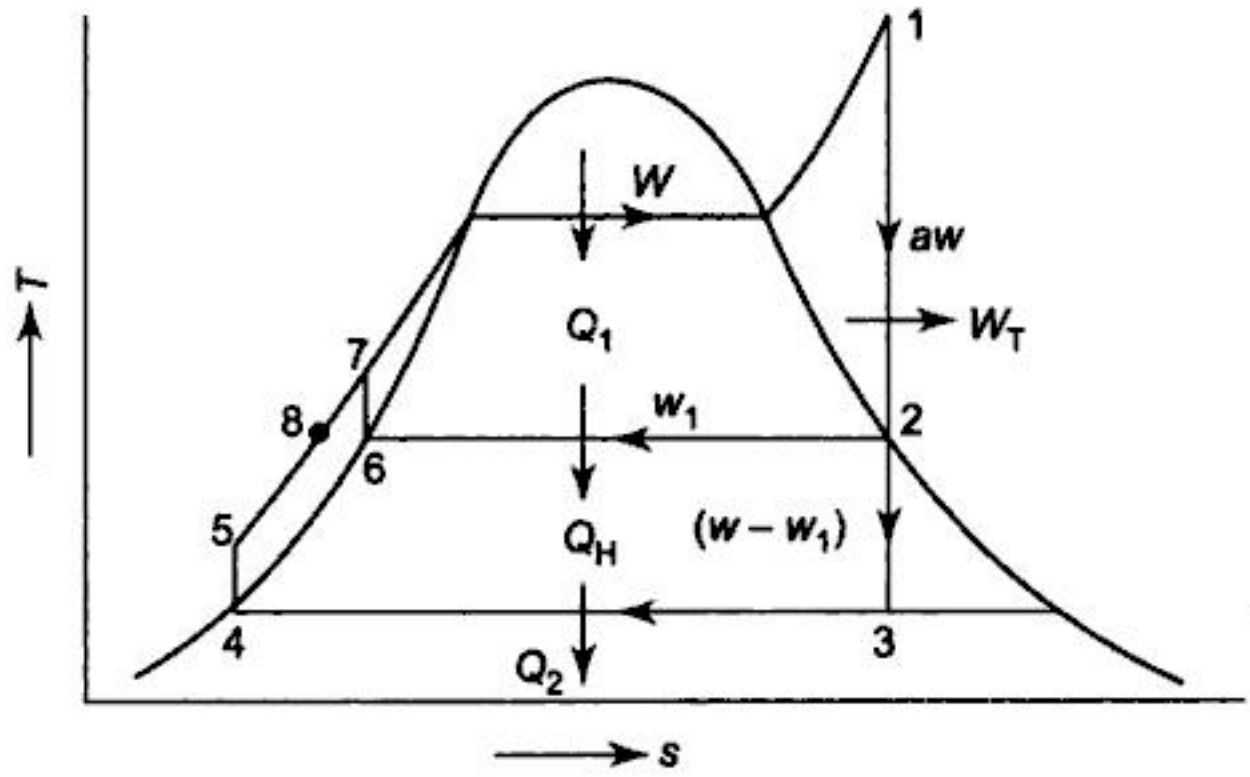

Fig. 12.36 *T-s Diagram of Power and Process Heat Plant*

$$Q_1 = w(h_1 - h_8)\text{kJ/h}$$

$$Q_2 = (w - w_1)\,(h_3 - h_4)\text{kJ/h}$$

$$Q_H = w_1(h_2 - h_6)\ \text{kJ/h}$$

$$W_T = w(h_1 - h_2) + (w - w_1)\,(h_2 - h_3)\ \text{kJ/h}$$

$$W_P = (w - w_1)\,(h_5 - h_4) + w_1(h_7 - h_6)\ \text{kJ/h}$$

$$w_1 h_7 + (w - w_1)h_5 = w \times h_8$$

where w is the boiler capacity (kg/h) and w_1 is the steam flow rate required (kg/h) at the desired temperature for process heating.

12.16 EFFICIENCIES IN STEAM POWER PLANT

For the steady flow operation of a turbine, neglecting changes in K.E. and P.E. (Figs. 12.37 and 12.38).

Maximum or ideal work output per unit mass of steam

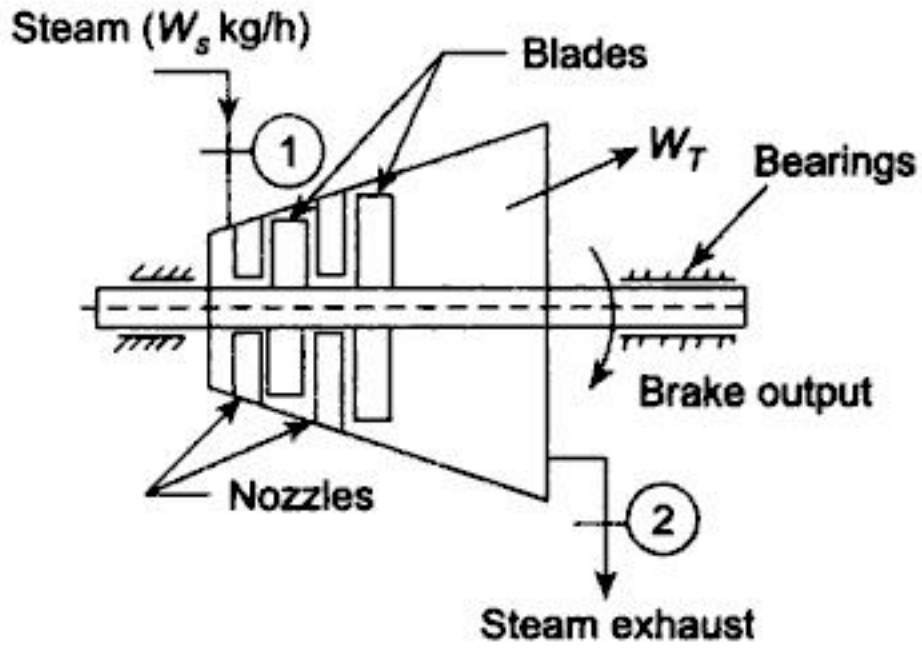

Fig. 12.37 *Efficiencies in a Steam Turbine*

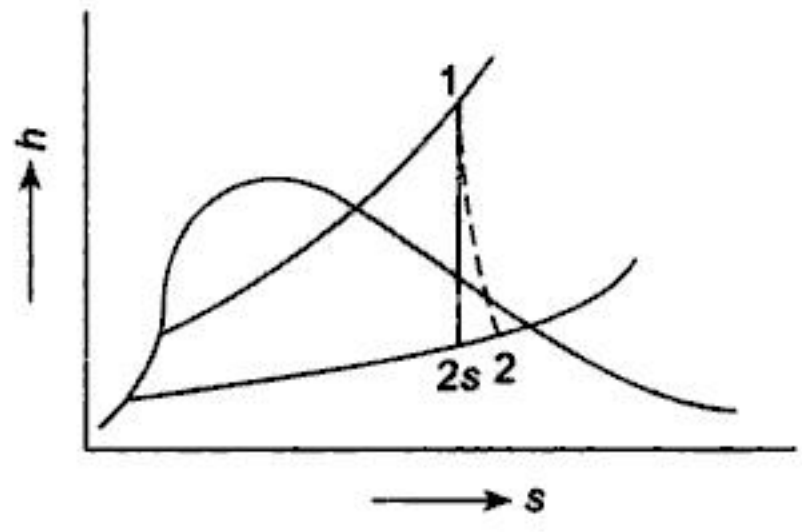

Fig. 12.38 *Internal Efficiency of a* ***Steam Turbine***

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$$\therefore \qquad \eta_{\text{cycle}} = \frac{W_{\text{net}}}{Q_1} = \frac{969.61}{2983.41} = \mathbf{0.325, \text{ or } 32.5\%}$$

(b) If $\eta_P = 80\%$, and $\eta_T = 80\%$

$$W_P = \frac{2.008}{0.8} = 2.51 \text{ kJ/kg}$$

$$W_T = 0.8 \times 971.62 = 777.3 \text{ kJ/kg}$$

$$\therefore \qquad W_{\text{net}} = W_T - W_P = 774.8 \text{ kJ/kg}$$

$\therefore$ % Reduction in work output

$$= \frac{969.61 - 774.8}{969.61} \times 100 = \mathbf{20.1\%}$$

$$h_{4s} = 173.88 + 2.51 = 176.39 \text{ kJ/kg}$$

$$\therefore \qquad Q_1 = 3159.3 - 176.39 = 2982.91 \text{ kJ/kg}$$

$$\therefore \qquad \eta_{\text{cycle}} = \frac{774.8}{2982.91} = 0.2597, \text{ or } 25.97\%$$

$\therefore$ % Reduction in cycle efficiency

$$= \frac{0.325 - 0.2597}{0.325} \times 100 = \mathbf{20.1\%}$$

Example 12.3 A cyclic steam power plant is to be designed for a steam temperature at turbine inlet of 360°C and an exhaust pressure of 0.08 bar. After isentropic expansion of steam in the turbine, the moisture content at the turbine exhaust is not to exceed 15%. Determine the greatest allowable steam pressure at the turbine inlet, and calculate the Rankine cycle efficiency for these steam conditions. Estimate also the mean temperature of heat addition.

Solution As state 2*s* (Fig. 12.42), the quality and pressure are known.

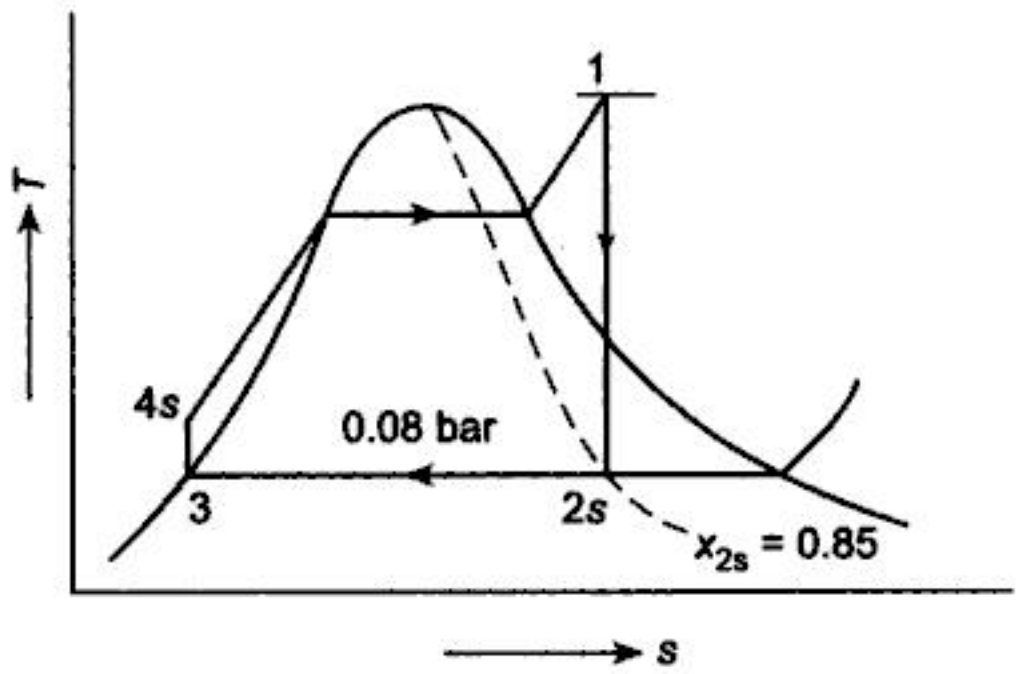

Fig. 12.42

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$$Q_1 = h_1 - h_6 = 3230.9 - 640.23 = 2590.67 \text{ kJ/kg}$$

$$\eta_{\text{cycle}} = \frac{916.13}{2590.67} = \textbf{0.3536, or 35.36\%}$$

$$\text{Steam rate} = \frac{3600}{916.13} = \textbf{3.93 kg/kW h}$$

$$T_{m1} = \frac{h_1 - h_7}{s_1 - s_7} = \frac{2590.67}{6.9212 - 1.8607} = 511.95 \text{ K} = 238.8°\text{C}$$

$$T_{m1} \text{ (without regeneration)} = \frac{h_1 - h_4}{s_1 - s_4}$$

$$= \frac{3039.07}{6.9212 - 0.6493}$$

$$= 484.55 \text{ K}$$

$$= 211.4 °\text{C}$$

Increase in T_{m1} due to regeneration

$$= 238.8 - 211.4 = \textbf{27.4°C}$$

$$W_T \text{ (without regeneration)} = h_1 - h_3$$

$$= 3230.9 - 2192.2 = 1038.7 \text{ kJ/kg}$$

$$\text{Steam rate (without regeneration)} = \frac{3600}{1038.7} = 3.46 \text{ kg/kW h}$$

$\therefore$ Increase in steam rate due to regeneration

$$= 3.93 - 3.46 = \textbf{0.476 kg/kW h}$$

$$\eta_{\text{cycle}} \text{ (without regeneration)} = \frac{h_1 - h_3}{h_1 - h_4} = \frac{1038.7}{3039.07}$$

$$= 0.3418 \text{ or } 34.18\%$$

$\therefore$ Increase in cycle efficiency due to regeneration

$$= 35.36 - 34.18 = \textbf{1.18\%}$$

Example 12.6 In a steam power plant the condition of steam at inlet to the steam generator is 20 bar and 300°C and the condenser pressure is 0.1 bar. Two feedwater heaters operate at optimum temperature. Determine: (a) the quality of steam at turbine exhaust, (b) net work per kg of steam, (c) cycle efficiency, and (d) the stream rate. Neglect pump work.

Solution From Fig. 12.19 (a),

$$h_1 = 3023.5 \text{ kJ/kg}$$

$$s_1 = 6.7664 \text{ kJ/kg K} = s_2 = s_3 = s_4$$

$$t_{\text{sat}} \text{ at } 20 \text{ bar} \cong 212°\text{C}$$

$$t_{\text{sat}} \text{ at } 0.1 \text{ bar} \cong 46°\text{C}$$

$$\Delta t_{\text{OA}} = 212 - 46 = 166°\text{C}$$

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$$= h_f + x_{2s}\, h_{fg}$$
$$= 503.71 + x_{2s} \times 2202.6$$

$$\therefore \quad x_{2s} = \frac{2123.89}{2202.6} = 0.964$$

$$\therefore \quad s_{2s} = s_f + x_{2s} \cdot s_{fg} = 1.5276 + 0.964 \times 5.6020$$
$$= 6.928 \text{ kJ/kg K}$$
$$= s_1$$

At state 1, $h_1 = 3152.3$ kJ/kg

$s_1 = 6.928$ kJ/kg K

From the Mollier chart

$p_1 = 22.5$ bar

$t_1 = 360°C$

Example 12.10 A certain factory has an average electrical load of 1500 kW and requires 3.5 MJ/s for heating purposes. It is proposed to install a single-extraction passout steam turbine to operate under the following conditions:

Initial pressure 15 bar.

Initial temperature 300°C.

Condenser pressure 0.1 bar.

Steam is extracted between the two turbine sections at 3 bar, 0.96 dry, and is isobarically cooled without subcooling in heaters to supply the heating load. The internal efficiency of the turbine (in the L.P. Section) is 0.80 and the efficiency of the boiler is 0.85 when using oil of calorific value 44 MJ/kg.

If 10% of boiler steam is used for auxiliaries calculate the oil consumption per day. Assume that the condensate from the heaters (at 3 bar) and that from the condenser (at 0.1 bar) mix freely in a separate vessel (hot well) before being pumped to the boiler. Neglect extraneous losses.

Solution Let w_s be the flow rate of steam (kg/h) entering the turbine, and w the amount of steam extracted per hour for process heat (Fig. 12.48).

$h_1 = 3037.3$ kJ/kg

$h_2 = 561.47 + 0.96 \times 2163.8$
$= 2638.7$ kJ/kg

$s_2 = 1.6718 + 0.96 \times 5.3201$
$= 6.7791$ kJ/kg K
$= s_{3s}$

$s_{3s} = 6.7791 = 0.6493 + x_{3s} \times 7.5009$

$$x_{3s} = \frac{6.1298}{7.5009} = 0.817$$

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$$(W_T)_m = h_a - h_b = 355.98 - 254.88$$
$$= 101.1 \text{ kJ/kg}$$
$$(W_P)_m = h_d - h_c = 76.5 \times 10^{-6} \times 4.46 \times 100 = 3.41 \times 10^{-2} \text{ kJ /kg}$$
$$\therefore \quad W_{\text{net}} = 101.1 \text{ kJ/kg}$$
$$Q_1 = h_a - h_d = 355.98 - 29.98 = 326 \text{ kJ/kg}$$
$$\therefore \quad \eta_m = \frac{W_{net}}{Q_1} = \frac{101.1}{326} = 0.31 \text{ or } 31\%$$

For the steam cycle

$$h_1 = 2792.2 \text{ kJ/kg}$$
$$s_1 = 6.4448 \text{ kJ/kg K} = s_2 = s_f + x_2\, s_{fg2}$$
$$= 0.4226 + x_2\,(8.4746 - 0.4226)$$
$$x_2 = \frac{6.0222}{8.0520} = 0.748$$
$$h_2 = 121.46 + 0.748 \times 2432.9 = 1941.27 \text{ kJ/kg}$$
$$(W_T)_{\text{St}} = h_1 - h_2 = 2792.2 - 1941.27 = 850.93 \text{ kJ/kg}$$
$$(W_P)_{\text{St}} = h_4 - h_3 = 0.001 \times 14.96 \times 100 = 1.496 \text{ kJ/kg} \cong 1.5 \text{ kJ/kg}$$
$$h_4 = 121.46 + 1.5 = 122.96 \text{ kJ/kg}$$
$$Q_1 = h_1 - h_4 = 2792.2 - 122.96 = 2669.24 \text{ kJ/kg}$$
$$(W_{\text{net}})_{St} = 850.93 - 1.5 = 849.43 \text{ kJ/kg}$$
$$\therefore \quad \eta_{\text{st}} = \frac{W_{\text{net}}}{Q_1} = \frac{849.43}{2669.24} = 0.318 \text{ or } 31.8\%$$

Overall efficiency of the binary cycle would be

$$\eta_{\text{overall}} = \eta_{\text{m}} + \eta_{\text{St}} - \eta_{\text{m}} \cdot \eta_{\text{St}}$$
$$= 0.31 + 0.318 - 0.31 \times 0.318$$
$$= \mathbf{0.5294 \text{ or } 52.94\%}$$

η_{overall} can also be determined in, the following way:

By writing the energy balance for a mercury condenser-steam boiler

$$m(h_b - h_c) = 1(h_1 - h_4)$$

where m is the amount of mercury circulating for 1 kg of steam in the bottom cycle.

$$\therefore \quad m = \frac{h_1 - h_4}{h_b - h_c} = \frac{2669.24}{254.88 - 29.98} = \frac{2669.24}{224.90} = 11.87 \text{ kg}$$
$$(Q_1)_{\text{total}} = m(h_a - h_d) = 11.87 \times 326 = 3869.6 \text{ kJ/kg}$$
$$(W_T)_{\text{total}} = m(h_a - h_b) + (h_1 - h_2)$$
$$= 11.87 \times 101.1 + 850.93 = 2051 \text{ kJ/kg}$$

$(W_P)_{\text{total}}$ may be neglected

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steam is bled for feedwater heating in a direct contact heater, which raises the feedwater to the saturation temperature of the bled steam. Calculate the mass of steam bled per kg of high pressure steam and the cycle efficiency. Assume that the L.P. expansion condition line is straight.

Ans. 0.53 kg; 31.9%

12.25 An ideal steam power plant operates between 70 bar, 550°C and 0.075 bar. It has seven feedwater heaters. Find the optimum pressure and temperature at which each of the heaters operate.

12.26 In a reheat cycle steam at 550°C expands in an h.p. turbine till it is saturated vapour. It is reheated at constant pressure to 400°C and then expands in a l.p. turbine to 40°C. If the moisture content at turbine exhaust is given to be 14.67%, find (a) the reheat pressure, (b) the pressure of steam at inlet to the h.p. turbine, (c) the net work output per kg, and (d) the cycle efficiency. Assume all processes to be ideal.

Ans. (a) 20 bar, (b) 200 bar, (c) 1604 kJ/kg, (d) 43.8%

12.27 In a reheat steam cycle, the maximum steam temperature is limited to 500°C. The condenser pressure is 0.1 bar and the quality at turbine exhaust is 0.8778. Had there been no reheat, the exhaust quality would have been 0.7592. Assuming ideal processes, determine (a) reheat pressure, (b) the boiler pressure, (c) the cycle efficiency, and (d) the steam rate.

Ans. (a) 30 bar, (b) 150 bar, (c) 50.51%, (d) 1.9412 kg/kWh

12.28 In a cogeneration plant, steam enters the h.p. stage of a two-stage turbine at 1 MPa, 200°C and leaves it at 0.3 MPa. At this point some of the steam is bled off and passed through a heat exchanger which it leaves as saturated liquid at 0.3 MPa. The remaining steam expands in the l.p. stage of the turbine to 40 kPa. The turbine is required to produce a total power of 1 MW and the heat exchanger to provide a heating rate of 500 KW. Calculate the required mass flow rate of steam into the h.p. stage of the turbine. Assume (a) steady condition throughout the plant, (b) velocity and gravity terms to be negligible, (c) both turbine stages are adiabatic with isentropic efficiencies of 0.80.

Ans. 2.457 kg/s

12.29 A steam power plant is designed to operate on the basic Rankine cycle. The heat input to the boiler is at the rate of 50 MW. The H_2O exits the condenser as saturated liquid and exits the boiler as saturated vapour. The pressure of steam at boiler exit is 120 bar and the condenser pressure is 0.04 bar. The heat input to the boiler is provided by a steady stream of hot gases initially at 2200 K and 1 atm. The hot gases exhaust at 600 K and 1 atm to the surroundings which are at 600 K and 1 atm. Taking the c_p of hot gases as 1.1 kJ/kgK, determine (a) the cycle efficiency, (b) the work output, (c) the power output (in MW), (d) the required mass flow rate of steam (in kg/h), (e) the specific steam consumption (in kg/k Wh), (f) the mass flow rate (in kg/h) of the stream of hot gases, (g) the exergy flux (in MW) of the inlet gases, (h) the exergy loss rate (in MW) with the exhaust gases, (i) the exergy consumption (in MW) in the steam generation process, (j) the exergy consumption (in MW) in the condensation process, (i) the second law efficiency.

Ans. (f) 1.20×10^5 kg/h, (g) 41.5 MW, (h) 3.14 W,
(i) 16.9 MW, (j) 1.40 MW, (i) 48.3%

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Due to heat transfers at constant volume processes, the efficiency of the Stirling cycle is less than that of the Carnot cycle. However, if a regenerative arrangement is used such that

$Q_{2-3} = Q_{4-1}$, i.e. the area under 2–3 is equal to the area under 4–1, then the cycle efficiency becomes

$$\eta = \frac{RT_1 \ln \frac{v_2}{v_1} - RT_2 \ln \frac{v_3}{v_4}}{RT_1 \ln \frac{v_2}{v_1}} = \frac{T_1 - T_2}{T_1} \tag{13.2}$$

So, the regenerative Stirling cycle has the same efficiency as the Carnot cycle.

13.3 ERICSSON CYCLE (1850)

The Ericsson cycle (Fig. 13.3) is made up of:

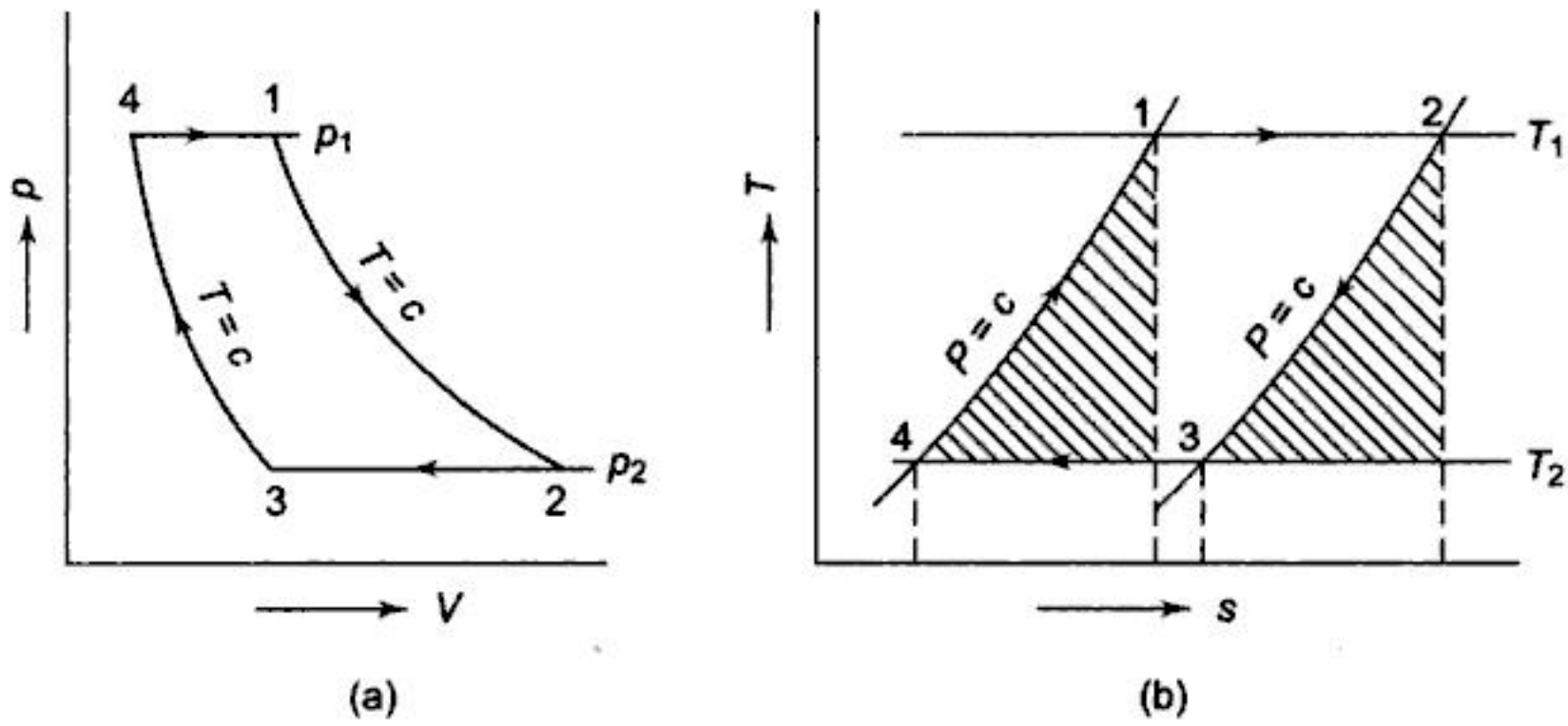

Fig. 13.3 *Ericsson Cycle*

Two reversible isotherms and two reversible isobars.

For 1 kg of ideal gas

$$Q_{1-2} = W_{1-2} = RT_1 \ln \frac{p_1}{p_2}$$

$$Q_{2-3} = c_p(T_2 - T_1);\ W_{2-3} = p_2(v_3 - v_2) = R(T_2 - T_1)$$

$$Q_{3-4} = W_{3-4} = -RT_2 \ln \frac{p_1}{p_2}$$

$$Q_{4-1} = c_p(T_1 - T_4);\ W_{4-1} = p_1(v_1 - v_4) = R(T_1 - T_2)$$

Since part of the heat is transferred at constant pressure and part at constant temperature, the efficiency of the Ericsson cycle is less than that of the Carnot cycle. But with ideal regeneration, $Q_{2-3} = Q_{4-1}$ so that all the heat is added from the external source at T_1 and all the heat is rejected to an external sink at T_2, the efficiency of the cycle becomes equal to the Carnot cycle efficiency, since

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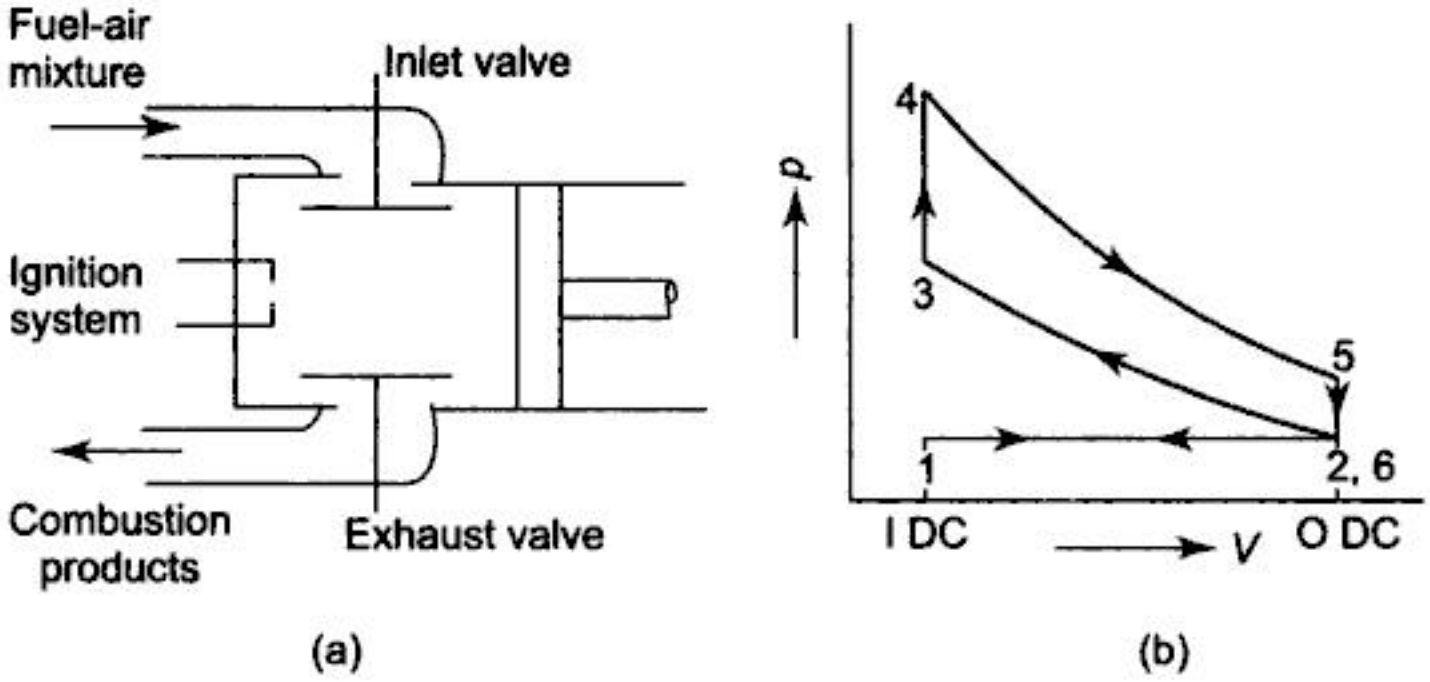

Fig. 13.5 *(a) S.I. Engine (Horizontal) (b) Indicator Diagram*

Process 1-2, Intake. The inlet valve is open, the piston moves to the right, admitting fuel-air mixture into the cylinder at constant pressure.
Process 2-3, Compression. Both the valves are closed, the piston compresses the combustible mixture to the minimum volume.
Process 3-4, Combustion. The mixture is then ignited by means of a spark, combustion takes place, and there is an increase in temperature and pressure.
Process 4-5, Expansion. The products of combustion do work on the piston which moves to the right, and the pressure and temperature of the gases decrease.
Process 5-6, Blow-down. The exhaust valve opens, and the pressure drops to the initial pressure.
Process 6-1, Exhaust. With the exhaust valve open, the piston moves inwards to expel the combustion products from the cylinder at constant pressure.

The series of processes as described above constitute a *mechanical cycle*, and not a thermodynamic cycle. The cycle is completed in four strokes of the piston.

Figure 13.5 (c) shows the air standard cycle (Otto cycle) corresponding to the above engine. It consists of: Two reversible adiabatics, one reversible isobar, and one reversible isochore.

Air is compressed in process 1-2 reversibly and adiabatically. Heat is then added to air reversibly at constant volume in process 2-3. Work is done by air in expanding reversibly and adiabatically in process 3-4. Heat is then rejected by air reversibly at constant volume in process 4-1, and the system (air) comes back to its initial state. Heat transfer processes have been substituted for the combustion and blow-down processes of the engine. The intake and exhaust processes of the engine cancel each other.

Let m be the fixed mass of air undergoing the cycle of operations as described above.

$$\text{Heat supplied } Q_1 = Q_{2-3} = mc_v (T_3 - T_2)$$

$$\text{Heat rejected } Q_2 = Q_{4-1} = mc_v (T_4 - T_1)$$

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$$T_3 = \frac{T_4}{r_c}$$

Process 2-3

$$\frac{p_2 V_2}{T_2} = \frac{p_3 V_3}{T_3}$$

$$T_2 = T_3 \frac{p_2}{p_3} = \frac{T_4}{r_p \cdot r_c}$$

Process 1-2

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1} = \frac{1}{r_k^{\gamma-1}}$$

$$\therefore \qquad T_1 = \frac{T_4}{r_p \cdot r_c \cdot r_k^{\gamma-1}}$$

Process 4-5

$$\frac{T_5}{T_4} = \left(\frac{v_4}{v_5}\right)^{\gamma-1} = \frac{1}{r_e^{\gamma-1}}$$

$$\therefore \qquad T_5 = T_4 \cdot \frac{r_c^{\gamma-1}}{r_k^{\gamma-1}}$$

Substituting the values of T_1, T_2, T_3, and T_5 in the expression of efficiency (Eq. 13.8).

$$\eta = 1 - \frac{T_4 \cdot \dfrac{r_c^{\gamma-1}}{r_k^{\gamma-1}} - \dfrac{T_4}{r_p \cdot r_c \cdot r_k^{\gamma-1}}}{\left(\dfrac{T_4}{r_c} - \dfrac{T_4}{r_p \cdot r_c}\right) + \gamma\left(T_4 - \dfrac{T_4}{r_c}\right)}$$

$$\therefore \qquad \eta_{\text{Dual}} = 1 - \frac{1}{r_k^{\gamma-1}} \frac{r_p \cdot r_c^{\gamma} - 1}{r_p - 1 + \gamma r_p (r_c - 1)} \qquad (13.11)$$

13.9 COMPARISON OF OTTO, DIESEL, AND DUAL CYCLES

The three cycles can be compared on the basis of either the same compression ratio or the same maximum pressure and temperature.

Figure 13.9 shows the comparison of Otto, Diesel, and Dual cycles for the same compression ratio and heat rejection. Here

1–2–6–5 —Otto cycle

1–2–7–5 —Diesel cycle

1–2–3–4–5 —Dual cycle

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or $$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} = \left(\frac{v_2}{v_1}\right)^{\gamma-1}$$

If r_k = compression ratio = v_1/v_2 the efficiency becomes (from Eq. 13.14)

$$\eta = 1 - \left(\frac{v_2}{v_1}\right)^{\gamma-1}$$

or $$\eta_{\text{Brayton}} = 1 - \frac{1}{r_k^{\gamma-1}} \tag{13.15}$$

If r_p = pressure ratio = p_2/p_1 the efficiency may be expressed in the following form also

$$\eta = 1 - \left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma}$$

or $$\eta_{\text{Brayton}} = 1 - \frac{1}{(r_p)^{(\gamma-1)/\gamma}} \tag{13.16}$$

The efficiency of the Brayton cycle, therefore, depends upon either the compression ratio or the pressure ratio. For the same compression ratio, the Brayton cycle efficiency is equal to the Otto cycle efficiency.

A closed cycle gas turbine plant (Fig. 13.15) is used in a gas-cooled nuclear reactor plant, where the source is a high temperature gas-cooled reactor (HTGR) supplying heat from nuclear fission directly to the working fluid (a gas).

13.12.1 Comparison between Brayton Cycle and Rankine Cycle

Both Rankine cycle and Brayton cycle consist of two reversible isobars and two reversible adiabatics (Fig. 13.16). While in Rankine cycle, the working fluid undergoes phase change, in Brayton cycle the working fluid always remains in the gaseous phase. Both the pump and the steam turbine in the case of Rankine cycle, and the compressor and the gas turbine in the case of Brayton cycle operate through the same pressure difference of p_1 and p_2. All are steady-flow machines and the work transfer is given by $-\int_{p_1}^{p_2} v\, dp$. For Brayton cycle, the average specific volume of air handled by the compressor is less than the same of gas in the gas turbine (since the gas temperature is much higher), the work done by the gas turbine is more than the work input to the compressor, so that there is W_{net} available to deliver. In the case of Rankine cycle, the specific volume of water in the pump is much less than that of the steam expanding in the steam turbine, so $W_T >> W_p$. Therefore, steam power plants are more popular than the gas turbine plants for electricity generation.

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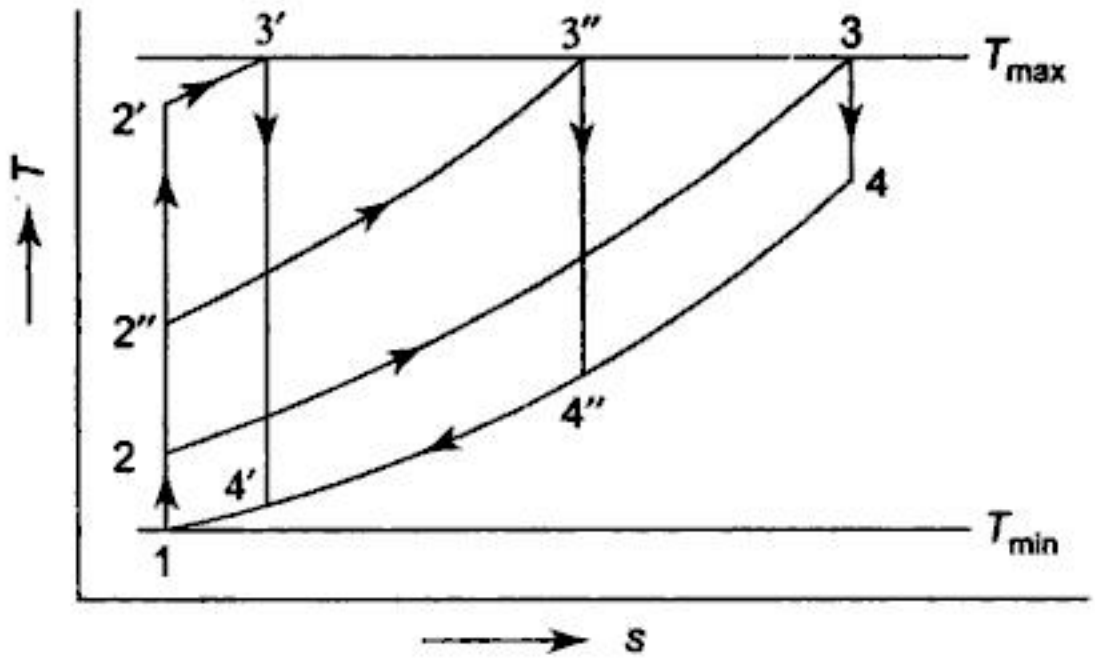

Fig. 13.20 *Effect of Pressure Ratio on Brayton Cycle*

As the pressure ratio is increased, the efficiency steadily increases, because T_{m1} increases and T_{m2} decreases. The mean temperature of heat addition T_{m1} approaches T_{max} and the mean temperature of heat rejection T_{m2} approaches T_{min}, with the increase in r_p. *In the limit when the compression process ends at T_{max}, the Carnot efficiency is reached, r_p has the maximum value (r_{pmax}), but the work capacity again become zero.*

Figure 13.21 shows how the cycle efficiency varies with the pressure ratio, with r_p varying between two limiting values of 1 and $(r_p)_{max}$ when the Carnot efficiency is reached. When $r_p = (r_p)_{max}$, the cycle efficiency is given by

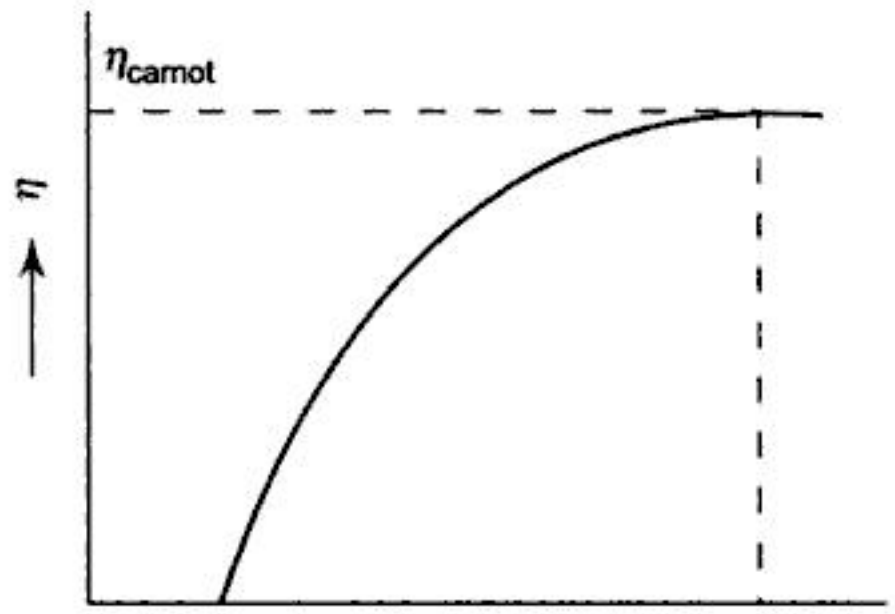

Fig. 13.21 *Effect of Pressure Ratio on Brayton Cycle Efficiency*

$$\eta = 1 - \frac{1}{((r_p)_{max}^{(\gamma-1)/\gamma}} = \eta_{Carnot}$$

$$= 1 - \frac{T_{min}}{T_{max}}$$

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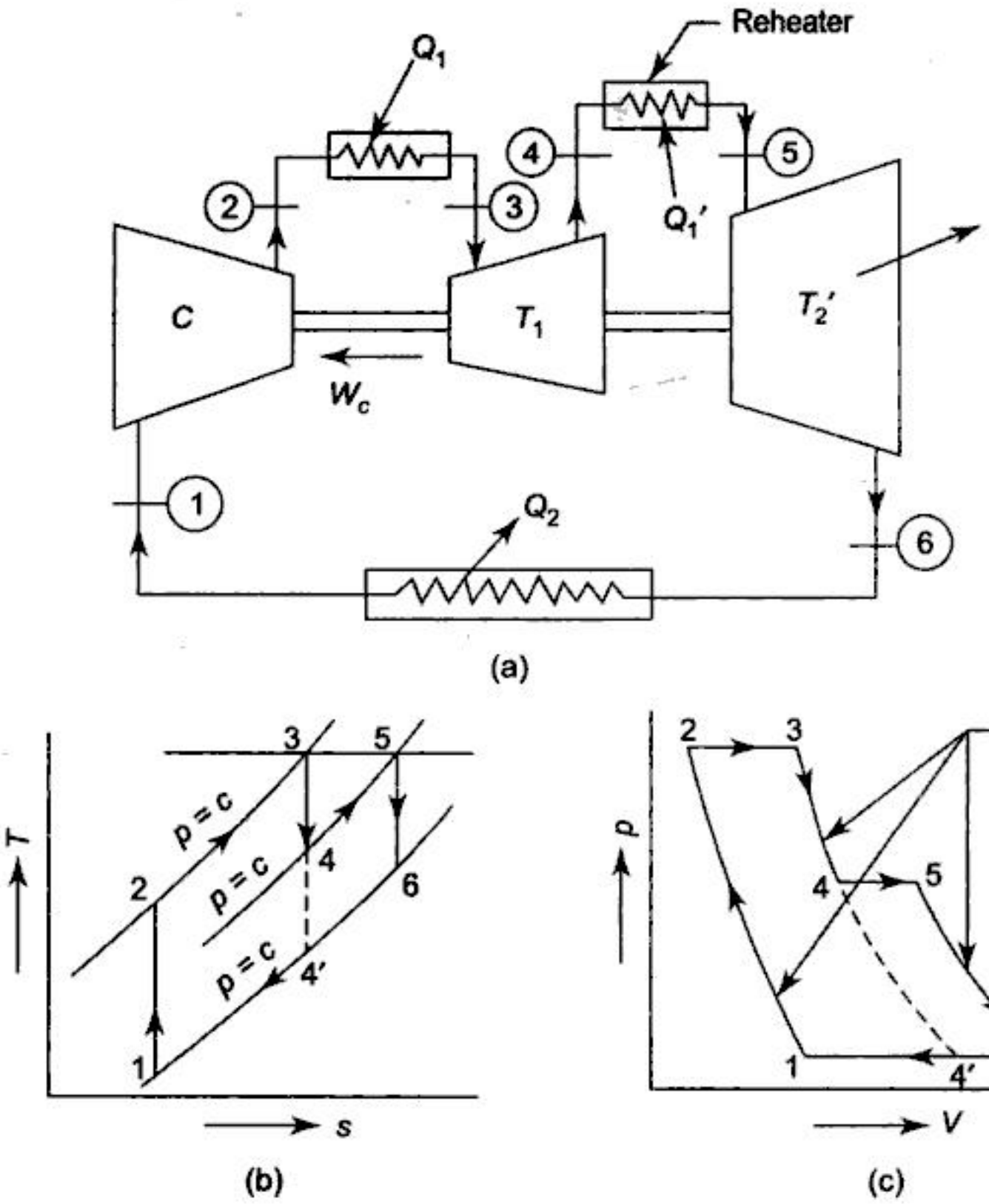

Fig. 13.24 *Effect of Reheat on Brayton Cycle*

If in one cycle, several stages of intercooling and several stages of reheat are employed, a cycle as shown in Fig. 13.25 is obtained. When the number of such stages is large the cycle reduces to the Ericsson cycle with two reversible isobars and two reversible isotherms. With ideal regeneration the cycle efficiency becomes equal to the Carnot efficiency.

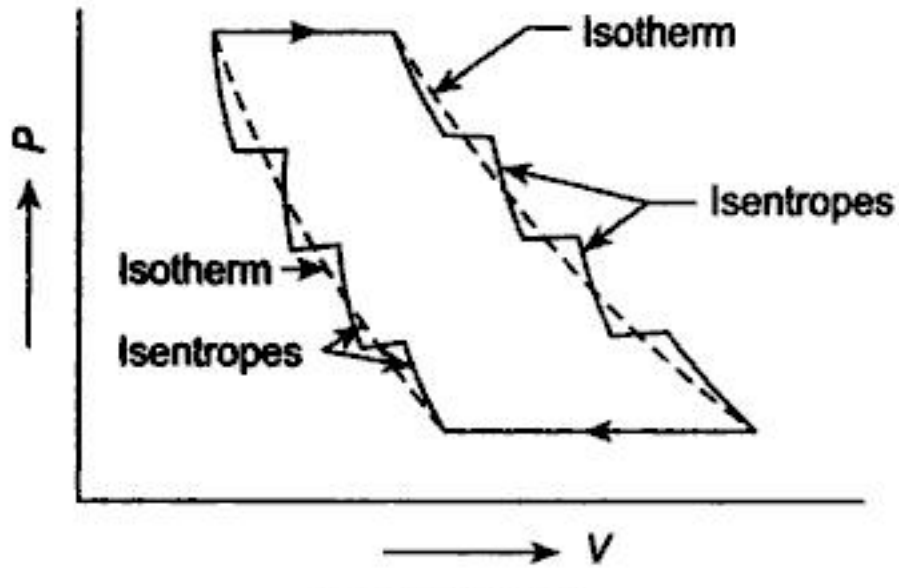

Fig. 13.25 (a)

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The power developed from the thrust of the engine is called the propulsive power, W_p, given by:

$$\dot{W}_p = F\overline{V}_{\text{aircraft}} = \dot{m}(\overline{V}_{\text{exit}} - \overline{V}_{\text{inlet}})\overline{V}_{\text{aircraft}} \tag{13.26}$$

The propulsive efficiency, η_p, is defined by:

$$\eta_p = \frac{\text{Propulsive power}}{\text{Energy input rate}} = \frac{\dot{W}_p}{\dot{Q}_{\text{in}}} \tag{13.27}$$

It is a measure of how efficiently the energy released during combustion is converted to propulsive power.

Space and weight limitations prohibit the use of regenerators and intercoolers on aircraft engines. The counterpart of reheating is afterburning. The air-fuel ratio in a jet engine is so high that the turbine exhaust gases are sufficiently rich in oxygen to support the combustion of more fuel in an afterburner (Fig. 13.29). Such burning of fuel raises the temperature of the gas before it expands in the nozzle, increasing the K.E. change in the nozzle and consequently increasing the thrust. In the air-standard case, the combustion is replaced by constant pressure heat addition.

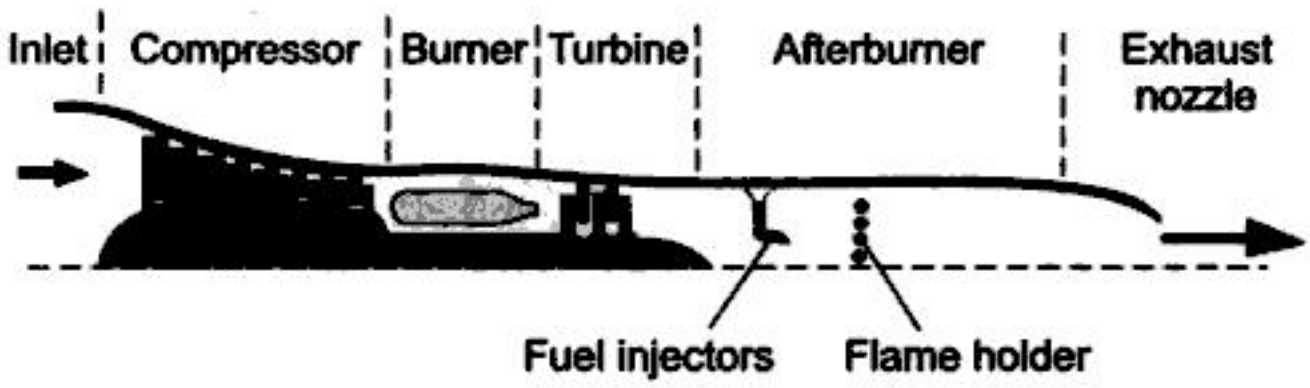

Fig. 13.29 *Turbojet Engine with Afterburner*

The most widely used engine in aircraft propulsion is the *turbofan engine* wherein a large fan driven by the turbine forces a considerable amount of air through a duct (cowl) surrounding the engine (Figs. 13.30 and 13.31). The fan

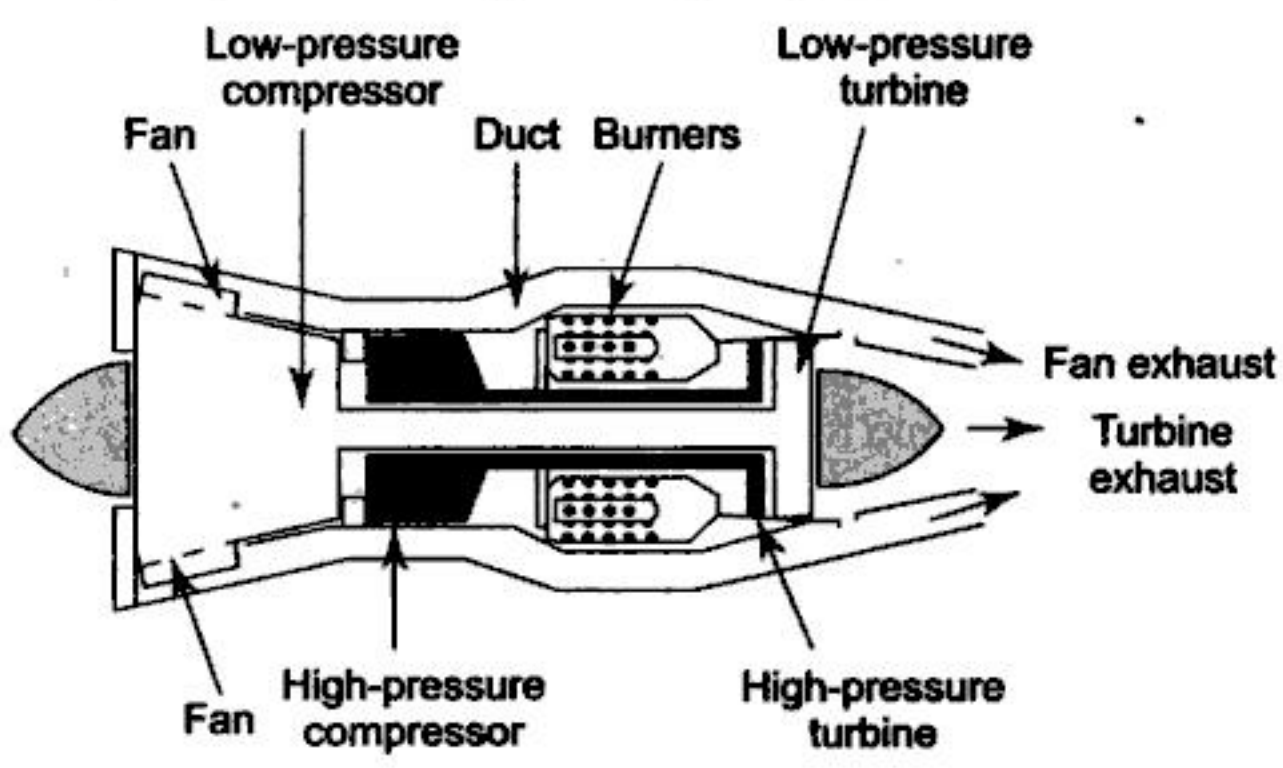

Fig. 13.30 *Turbofan Engine*

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(b) Heat supplied, $Q_1 = c_p (T_3 - T_2)$

$= 1.005\ (1753 - 873)$

$= 884.4$ kJ/kg

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{v_1}{v_2} \times \frac{v_2}{v_3}\right)^{\gamma-1} = \left(\frac{16}{2.01}\right)^{0.4} = 2.29$$

$$\therefore \quad T_4 = \frac{1753}{2.29} = 766 \text{ K}$$

Heat rejected, $Q_2 = c_v (T_4 - T_1) = 0.718\ (766 - 288) = 343.2$ kJ/kg

(c) $$\text{Cycle efficiency} = 1 - \frac{Q_2}{Q_1}$$

$$= 1 - \frac{343.2}{884.4} = \mathbf{0.612 \text{ or } 61.2\%}$$

It may also be estimated from the equation

$$\eta_{\text{cycle}} = 1 - \frac{1}{\gamma} \cdot \frac{1}{r_k^{\gamma-1}} \cdot \frac{r_c^{\gamma} - 1}{r_c - 1}$$

$$= 1 - \frac{1}{1.4} \frac{1}{(16)^{0.4}} \cdot \frac{(2.01)^{1.4} - 1}{2.01 - 1}$$

$$= 1 - \frac{1}{1.4} \cdot \frac{1}{3.03} \cdot 1.64 = \mathbf{0.612 \text{ or } 61.2\%}$$

$$W_{\text{net}} = Q_1 \times \eta_{\text{cycle}}$$

$$= 884.4 \times 0.612 = 541.3 \text{ kJ/kg}$$

$$v_1 = \frac{RT_1}{p_1} = \frac{0.287 \times 288}{100} = 0.827 \text{ m}^3/\text{kg}$$

$$v_2 = \frac{0.827}{16} = 0.052 \text{ m}^3/\text{kg}$$

$$\therefore \quad v_1 - v_2 = 0.827 - 0.052 = 0.775 \text{ m}^3/\text{kg}$$

(d) $$\text{m.e.p.} = \frac{W_{\text{net}}}{v_1 - v_2} = \frac{541.3}{0.775} = \mathbf{698.45 \text{ kPa}}$$

Example 13.4 An air standard dual cycle has a compression ratio of 16, and compression begins at 1 bar, 50°C. The maximum pressure is 70 bar. The heat transferred to air at constant pressure is equal to that at constant volume. Estimate (a) the pressures and temperatures at the cardinal points of the cycle, (b) the cycle efficiency, and (c) the m.e.p. of the cycle, $c_v = 0.718$ kJ/kg K, $c_p = 1.005$ kJ/kg K.

Solution Given: (Fig. 13.38)

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The theoretical yield, assuming perfect insulation, can be determined by a mass and energy balance for the control volume (Joule-Kelvin refrigeration system) as shown in Fig. 14.24. Let $\dot{m}_f$ be the rate at which liquid air is produced (the same given to the system as make-up), and $\dot{m}$ the rate at which air is compressed and then expanded. Then the yield is

$$Y = \frac{\dot{m}_f}{\dot{m}}$$

and the energy balance gives

$$\dot{m}h_2 - \dot{m}_f h_5 - (\dot{m} - \dot{m}_f)h_7 = 0$$

$$\therefore \quad \dot{m}(h_2 - h_7) - \dot{m}_f (h_5 - h_7) = 0$$

$$Y = \frac{\dot{m}_f}{\dot{m}} = \frac{h_2 - h_7}{h_5 - h_7}$$

$$\therefore \quad Y = \frac{h_7 - h_2}{h_7 - h_5} \tag{14.6}$$

No yield is thus possible unless h_7 is greater than h_2. The energy balance for the compressor gives

$$\dot{m}h_1 + W_c = \dot{m}h_2 + Q_R$$

where Q_R is the heat loss to the surroundings from the compressor

$$\therefore \quad \frac{W_c}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2)$$

This is the *minimum work requirement.*

Specific work consumption, W

$$= \frac{W_c}{\dot{m}} \times \frac{\dot{m}}{\dot{m}_f} = \frac{W_c}{\dot{m}} \frac{1}{Y} = \frac{h_7 - h_5}{h_7 - h_2} \; [T_1(s_1 - s_2) - (h_1 - h_2)]$$

14.7.2 Claude System of Air Liquefaction

In the Claude system, energy is removed from the gas stream by allowing it to do some work in an expander. The flow and T-s diagrams are given in Fig. 14.25.

The gas is first compressed to pressures of about 40 atm and then passed through the first heat exchanger. Approximatly 80% of the gas is then diverted from the main stream, expanded through an expander, and reunited with the return stream below the second heat exchanger. The stream to be liquefied continues through the second and third heat exchangers, and is finally expanded through an expansion valve to the liquid receiver. The cold vapour from the liquid receiver is returned through the heat exchangers to cool the incoming gas.

The yield and the specific work consumption may be computed by making the mass and energy balance as in the Linde-Hampson system.

14.8 PRODUCTION OF SOLID ICE

Dry ice is used for low temperature refrigeration, such as to preserve ice cream and other perishables. The property diagram of CO_2 on the p-h coordinates is given in Fig. 14.26. The schematic diagram of producing solid CO_2 and the corresponding p-h diagram are shown in Fig. 14.27 and Fig. 14.28 respectively.

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superheated in the suction line heat exchanger to the extent that a discharge temperature of 60°C is obtained after isentropic compression. Determine (a) the subcooling achieved in the heat exchanger, (b) the refrigerant flow rate in kg/s, (c) the cylinder dimensions of the two-cylinder compressor, if the speed is 900 rpm, stroke-to-bore ratio is 1.1, and the volumetric efficiency is 80%, (d) the COP of the plant, and (e) the power required to drive the compressor in kW.

Solution From the *p-h* chart of R-12, the property values at the states, as shown in Fig. 14.32.

$$h_3 = 882,\ h_2 = 1034$$

$$h_6 = 998,\ h_1 = 1008 \text{ kJ/kg}$$

$$v_1 = 0.084 \text{ m}^3/\text{kg}$$

$$h_3 - h_4 = h_1 - h_6$$

$$882 - h_4 = 1008 - 998 = 10$$

$$\therefore \quad h_4 = 872 \text{ kJ/kg}$$

$$\therefore \quad t_4 = 25°\text{C}$$

So 10°C subcooling is achieved in the heat exchanger. Refrigeration effect = $h_6 - h_5 = 998 - 872 = \mathbf{126 \text{ kJ/kg}}$

$$\therefore \quad \text{Refrigerant flow rate } \frac{10 \times 14000}{126} = \mathbf{1110 \text{ kg/h} = 0.31 \text{ kg/s}}$$

$$\text{Volume flow rate} = w \cdot v_1 = 1110 \times 0.084 = 93 \text{ m}^3/\text{h}$$

$$\text{Compressor displacement} = \frac{93}{0.8} = 116 \text{ m}^3/\text{h} = 1.94 \text{ m}^3/\text{min}$$

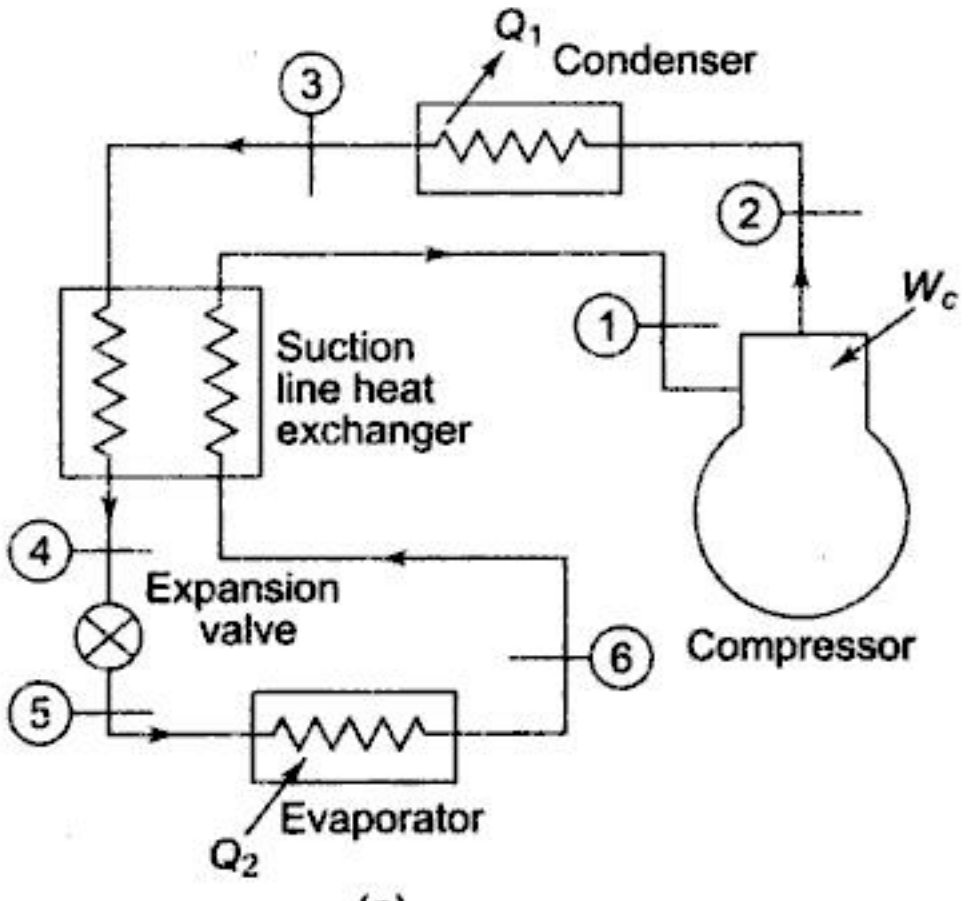

Fig. 14.32

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Table 16.3 *Fundamental Ideal Gas Reactions*

Reaction	ΔH_0, *J/gmol*	ΔS_0 *J/gmol-K*	ΔG°_{298} *J/gmol*
$H_2 \rightleftharpoons 2H$	427,380	4.90	404,335
$HCl \rightleftharpoons \frac{1}{2}H_2 + \frac{1}{2}Cl_2$	91,760	– 22.25	95.110
$HBr \rightleftharpoons \frac{1}{2}H_2 + \frac{1}{2}Br_2$	50,280	– 24.05	54,050
$HI \rightleftharpoons \frac{1}{2}H_2 + \frac{1}{2}I_2$	5,320	– 21.00	8,380
$NO \rightleftharpoons \frac{1}{2}N_2 + \frac{1}{2}O_2$	– 90,500	– 10.48	– 87,570
$H_2O \rightleftharpoons H_2 + \frac{1}{2}O_2$	239,250	14.70	232,545
$H_2S \rightleftharpoons H_2 + \frac{1}{2}S_2$	80,450	6.90	71,230
$CO_2 \rightleftharpoons CO + \frac{1}{2}O_2$	279,890	18.69	258,940
$NO_2 \rightleftharpoons NO + \frac{1}{2}O_2$	59,500	11.44	37,290
$SO_2 \rightleftharpoons \frac{1}{2}S_2 + O_2$	349,025	3.77	329,330
$NH_3 \rightleftharpoons \frac{1}{2}N_2 + \frac{3}{2}H_2$	39,800	40.27	16,380
$SO_3 \rightleftharpoons SO_2 + \frac{1}{2}O_2$	94,690	89.50	67,880
$N_2O_4 \rightleftharpoons 2NO_2$	56,980	174.30	5,030

Both ΔS_0 and ΔG°_{298} may be added and subtracted in the same manner as ΔH_0. For example.

$$H_2O \rightleftharpoons H_2 + \frac{1}{2}O_2 \qquad \Delta G^\circ_{298} = 232{,}545 \text{ J/gmol}$$

$$CO_2 \rightleftharpoons CO + \frac{1}{2}O_2 \qquad \Delta G^\circ_{298} = 258{,}940 \text{ J/gmol}$$

$$CO + H_2O \rightleftharpoons CO_2 + H_2 \qquad \Delta G^\circ_{298} = -26{,}395 \text{ J/gmol}$$

$$\ln K_{298} = \frac{\Delta G^\circ_{298}}{\bar{R}T} = \frac{26{,}395}{8.3143 \times 298} = 10.653$$

$$K_{298} = 42{,}330$$

From this, the value of the degree of reaction at equilibrium ε_e may be calculated.

From Eq. (16.3), if $n_0 = 1$ and $n'_0 = 0$

$$\left(\frac{\partial G}{\partial \varepsilon}\right)_{T,p} = (\nu_3\mu_3 + \nu_4\mu_4 - \nu_1\mu_1 - \nu_2\mu_2)$$

Since $$\mu_k = \bar{R}T\ (\phi_k + \ln p + \ln x_k)$$

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16.14 FIRST LAW FOR REACTIVE SYSTEMS

For the S.S.S.F. process as shown in Fig. 16.4 the first law gives

$$H_R + Q_{C.V.} = H_P + W_{C.V.}$$

or
$$\sum_R n_i \bar{h}_i + Q_{C.V.} = \sum_P n_e \bar{h}_e + W_{C.V.}$$

$$\sum_R n_i \left[\bar{h}_f^0 + \Delta\bar{h}\right]_i + Q_{C.V.} = \sum_P n_e \left[\bar{h}_f^0 + \Delta\bar{h}\right]_e + W_{C.V.} \qquad (16.19)$$

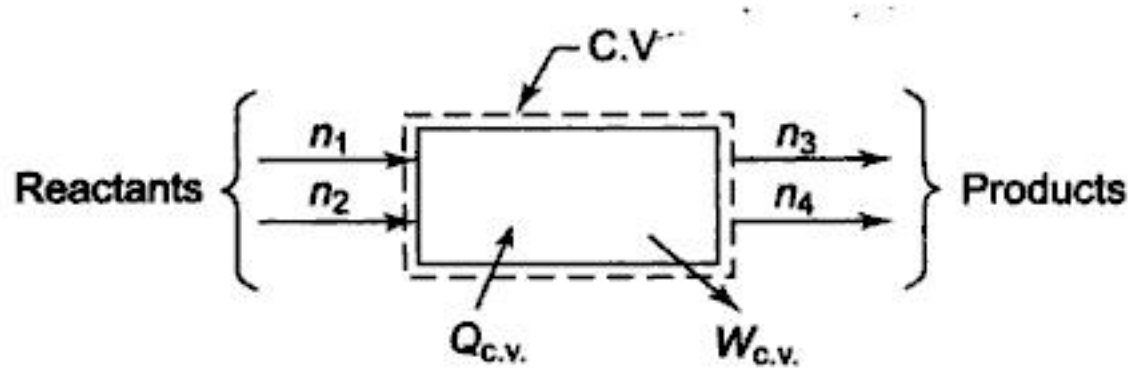

Fig. 16.4 *First law for a Reactive System*

Table 16.4 *Enthalpy of Formation, Gibbs Function of Formation, and Absolute Entropy of Various Substances at 25°C, 1 atm. pressure*

Substance	*Molecular Weight, M*	$\bar{h}_f^0$ *kJ/kg mol*	$\bar{g}_f^0$ *kJ/kg mol*	s^0 *kJ/kg mol K*
$CO(g)$	28.011	− 110529	− 137150	197.653
$CO_2(g)$	44.001	− 393522	− 394374	213.795
$H_2O(g)$	18.015	− 241827	− 228583	188.833
$H_2O(l)$	18.015	− 285838	− 237178	70.049
$CH_4(g)$	16.043	− 74873	− 50751	186.256
$C_2H_2(g)$	26.038	+ 226731	+ 209234	200.958
$C_2H_4(g)$	28.054	+ 52283	+ 68207	219.548
$C_2H_6(g)$	30.070	− 84667	− 32777	229.602
$C_3H_8(g)$	44.097	− 103847	− 23316	270.019
$C_4H_{10}(g)$	58.124	− 126148	− 16914	310.227
$C_8H_{18}(g)$	114.23	− 208447	+ 16859	466.835
$C_8H_{18}(l)$	114.23	− 249952	+ 6940	360.896

When the states of reactants and products are not in the standard reference state (298K, 1 atm), then, as shown in Fig. 16.5,

$$Q_{C.V.} = H_P - H_R = (H_P - H_{P0}) + (H_{P0} - H_{R0}) + (H_{R0} - H_R)$$

$$= \sum_p n_e \left(\bar{h}_p - \bar{h}_{p0}\right) + \Delta\bar{h}^{\,0}_{RP} - \sum_R n_i \left(\bar{h}_R - \bar{h}_{R0}\right) \qquad (16.20)$$

where $\Delta\bar{h}^{\,0}_{PR}$ is the enthalpy of reaction at the standard temperature (298 K). The variation of enthalpy with pressure is not significant.

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$$W_{rev} = (\bar{g}_{f0})_{C_3H_8(g)} - 3(\bar{g}_{f0})_{CO_2} - 4(\bar{g}_{f0})_{H_2O(g)}$$
$$= -23{,}316 - 3(-394{,}374) - 4(-228{,}583)$$
$$= \mathbf{2{,}074{,}128\ kJ/kg\ mol}$$
$$= \frac{2{,}074{,}128}{44.097} = \mathbf{47{,}035.6\ kJ/kg}$$

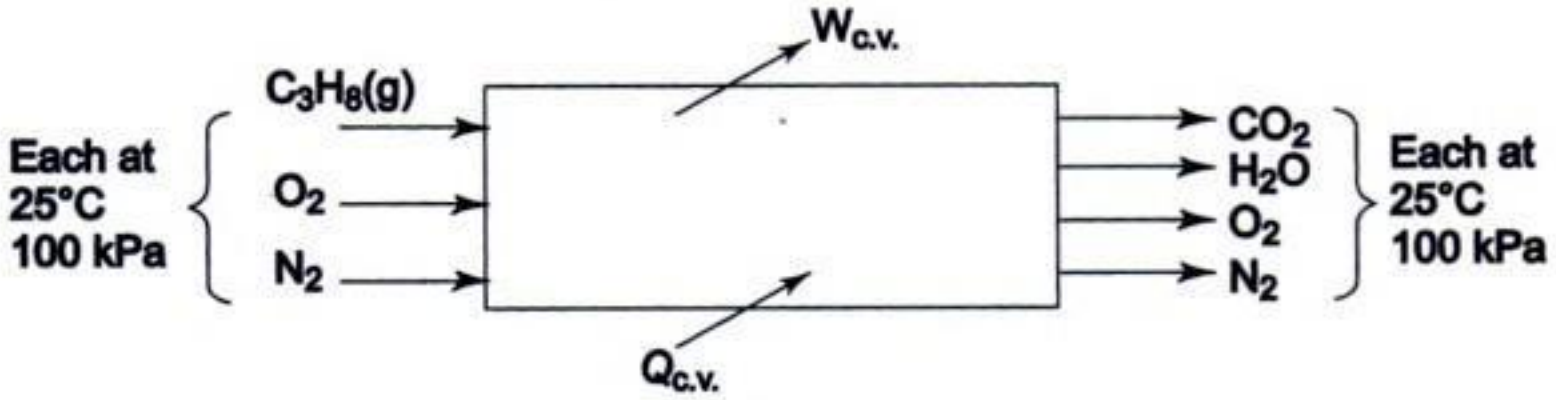

Fig. 16.10

(b) $H_R = H_P$

$$(\bar{h}_{f0})_{C_3H_8(g)} = 3(\bar{h}_{f0} + \Delta\bar{h})_{CO_2} + 4(\bar{h}_{f0} + \Delta\bar{h})_{H_2O(g)} + 15\Delta\bar{h}_{O_2} + 75.2\Delta\bar{h}_{N_2}$$

From Table 16.4

$$-103{,}847 = 3(-393{,}522 + \Delta\bar{h})_{CO_2} + 4(-241{,}827 + \Delta\bar{h})_{H_2O}(g) + 15\,\Delta\bar{h}_{O_2} + 75.2\,\Delta\bar{h}_{N_2}$$

Using Table C in the appendix, and by trial and error, the adiabatic flame temperature is found to be 980 K.

The entropy of the reactants

$$S_R = \sum_R (n_i\bar{s}_i^0)_{298} = (\bar{s}^0_{C_3H_8(g)} + 20\bar{s}^0_{O_2} + 75.2\bar{s}^0_{N_2})_{298}$$
$$= 270.019 + 20\ (205.142) + 75.2(191.611)$$
$$= 18{,}782.01\ \text{kJ/kg mol K}$$

The entropy of the products

$$S_P = \sum_P (n_e\bar{s}_e^0)_{980} = (3\bar{s}^0_{CO_2} + 4\bar{s}^0{}_{H_2O(g)} + 15\bar{s}^0_{O_2} + 75.2\bar{s}^0_{N_2})_{980}$$
$$= 3(268.194) + 4(231.849) + 15(242.855) + 75.2(227.485)$$
$$= 22{,}481.68\ \text{kJ/kg mol K}$$

$\therefore$ The increase in entropy during combustion

$$S_P - S_R = \mathbf{3{,}699.67\ kJ/kg\ mol\ K}$$

The irreversibility of the process

$$I = T_0\,[\sum_P n_e\bar{s}_e - \sum_R n_i\bar{s}_i]$$
$$= 298 \times 3699.67$$

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p. At a temperature of 3800 K, $K = 1$ atm and $\frac{\Delta H}{\overline{R}} = 59$ K, calculate the value of ε and $\left(\frac{\partial \varepsilon_e}{\partial T}\right)_p$ at a pressure of 1 atm.

16.10 (a) Starting with n_0 moles of water vapour which dissociates according to the equation $H_2O \rightleftharpoons H_2 + \frac{1}{2} O_2$, show that at equilibrium

$$K = \frac{\varepsilon_e^{3/2}}{(2+\varepsilon_e)^{1/2}(1-\varepsilon_e)} \cdot p^{1/2}$$

(b) At an average temperature of 1900 K, the slope of the graph of log K against $1/T$ for the dissociation of water vapour is found to be – 13,000. Determine the heat of dissociation. Is it exothermic or endothermic?

Ans. 248,885 kJ/kg mol, endothermic

16.11 The volumetric composition of the 'dry' products of combustion of an unknown hydrocarbon fuel, $C_x H_y$, gives: CO_2 12.1%, O_2 3.8%, CO 0.9% and N_2 83.2%. Detrmine (a) the chemical formula of the fuel, (b) the air fuel ratio, and (c) the percentage of excess air used.

Ans. (a) $C_{13} H_{23}$, (b) 17.03, (c) 18%

16.12 Liquid octane is burned with air in a combustion test, and the dry volumetric analysis of the products reveals the following composition: CO_2 11%, O_2 3.6%, CO 1.5% and N_2 83.9%. Compute the actual air-fuel ratio used during the test and the percentage excess air.

Ans. 17.267, 14.19%

16.13 Propane (C_3H_8) is reacted with air in such a ratio that an analysis of the products of combustion gives CO_2 11.5%, O_2 2.7% and CO 0.7%. What is the percent theoretical air used during the test?

Ans. 111%

16.14 Carbon monoxide and 300% theoretical air enter a steady flow combustor at 400K and a low pressure. The energy released by the reaction heats the product gases to 1400 K. If the combustion is complete, estimate the heat gained or lost through the walls of the combustor.

Ans. – 17,360 kJ/kg mol

16.15 Liquid octane enters the combustion chamber of a gas turbine at 25°C and air enters from the compressor at 227°C. It is determined that 98% of the carbon in the fuel burns to form CO_2 and the remaining 2% burns to form CO. The temperature of the products is limited to 827°C. Estimate the air-fuel ratio used and the percentage excess air.

Ans. 66,338%

16.16 A mixture of methane and oxygen, in the proper ratio for complete combustion and at 25°C and 1 atm, reacts in a constant volume calorimeter bomb. Heat is transferred until the products of combustion are at 400 K. Determine the heat transfer per mole of methane.

Ans. – 794414 kJ/kg mol

16.17 Liquid hydrazine (N_2H_4) and oxygen gas, both at 25°C, 0.1 MPa are fed to a rocket combustion chamber in the ratio of 0.5 kg O_2/kg N_2H_4. The heat transfer

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$$c = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} \tag{17.3}$$

No fluid is truly incompressible, although liquids show little change in density. The velocity of sound in common liquids is of the order of 1650 m/s.

17.1.1 Velocity of Sound in an Ideal Gas

For an ideal gas, in an isentropic process

$$pv^{\gamma} = \text{constant}$$

or

$$\frac{p}{\rho^{\gamma}} = \text{constant}$$

By logarithmic differentiation (i.e., first taking logarithm and then differentiating)

$$\frac{dp}{p} - \gamma \frac{d\rho}{\rho} = 0$$

$$\therefore \quad \frac{dp}{d\rho} = \gamma \frac{p}{\rho}$$

Since $c^2 = \frac{dp}{d\rho}$ and $p = \rho RT$

$$c^2 = \gamma RT$$

or

$$c = \sqrt{\gamma RT} \tag{17.4}$$

where R = characteristic gas constant = $\dfrac{\text{Universal gas constant}}{\text{Molecular weight}}$

The lower the molecular weight of the fluid and higher the value of γ, the higher is the sonic velocity at the same temperature. c is a thermodynamic property of the fluid.

17.1.2 Mach Number

The Mach number, M, is defined as the ratio of the actual velocity $\mathbf{V}$ to the sonic velocity c.

$$\mathrm{M} = \frac{\mathbf{V}}{c}$$

When $M > 1$, the flow is supersonic, when $M < 1$, the flow is subsonic, and when $M = 1$, the flow is sonic.

17.2 STAGNATION PROPERTIES

The isentropic *stagnation state* is defined as the state a fluid in motion would reach if it were brought to rest isentropically in a steady-flow, adiabatic, zero work output device. This is a reference state in a compressible fluid flow and is

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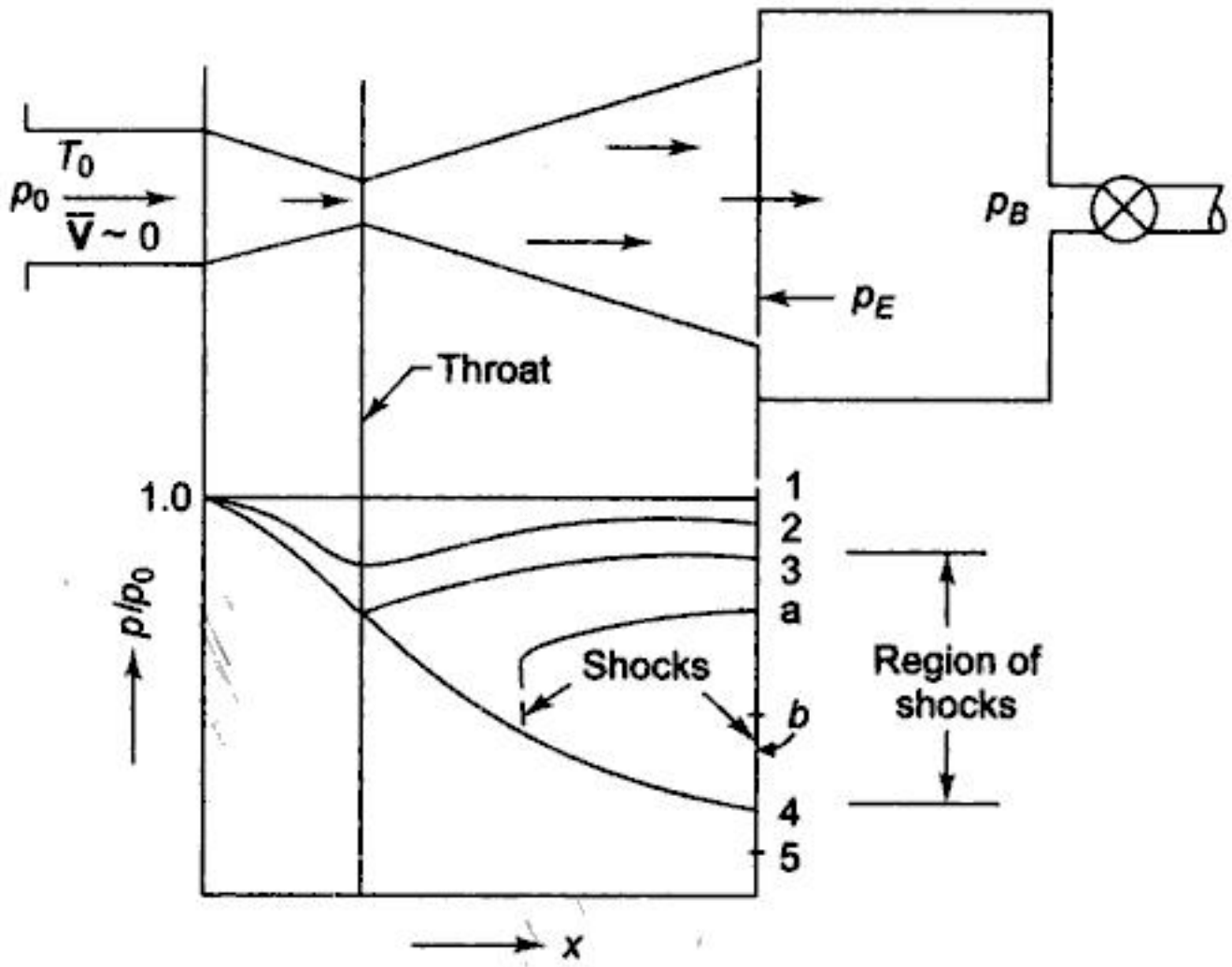

Fig. 17.8 *Compressible Flow Through a Convergent Divergent Nozzle*

than the critical pressure ratio, the velocity increases in the convergent section, but $M < 1$ at the throat. The divergent section acts as a subsonic diffuser in which the pressure increases and velocity decreases. Point 3 indicates the back pressure at which $M = 1$ at the throat, but the diverging section acts as a subsonic diffuser in which the pressure increases and velocity decreases. Point 4 indicates one other back pressure for which the flow is isentropic throughout and the diverging section acts as a supersonic nozzle with a continuous decrease in pressure and a continuous increase in velocity, and $p_{E4} = p_{B4}$. This condition of supersonic flow past the throat with the isentropic conditions is called the *design pressure ratio of the nozzle*. If the pressure is lowered to 5, no further decrease in exit pressure occurs and the drop in pressure from p_E to p_B occurs outside the nozzle.

Between the back pressures designated by points 3 and 4, flow is not isentropic in the diverging part, and it is accompanied by a highly irreversible phenomenon, known as *shocks*. Shocks occur only when the flow is supersonic, and after the shock the flow becomes subsonic, when the rest of the diverging portion acts as a diffuser. Properties vary discontinuously across the shock. When the back pressure is as indicated by point b (Fig. 17.8), the flow throughout the nozzle is isentropic, with pressure continuously decreasing and velocity increasing, but a shock appears just at the exit of the nozzle. When the back pressure is increased from b to a, the shock moves upstream, as indicated. When the back pressure is further increased, the shock moves further upstream and disappears at the nozzle throat where the back pressure corresponds to 3. Since flow throughout is subsonic, no shock is possible.

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$$\frac{T_y}{T_x} = 1.483,\ \frac{p_{oy}}{p_{ox}} = 0.84,\ M_y = 0.631$$

$$T_{ox} = T_{oy} = 573\ \text{K}$$

$$T_{ox} = \left(1 + \frac{\gamma - 1}{2} M_x^2\right) T_x = (1 + 0.2 \times 3) T_x = 1.6\, T_x$$

$$T_x = \frac{573}{1.6} = 358\ \text{K}$$

$$T_y = 358 \times 1.483 = 530\ \text{K} = 257°\text{C}$$

$$p_{ox} = \frac{p_{oy}}{0.84} = \frac{70}{0.84} = 83.3\ \text{kPa}$$

$$M_x = 1.735$$

Example 17.5 A convergent-divergent nozzle operates at off-design condition while conducting air from a high pressure tank to a large container. A normal shock occurs in the divergent part of the nozzle at a section where the cross-sectional area is 18.75 cm^2. If the stagnation pressure and stagnation temperature at the inlet of the nozzle are 0.21 MPa and 36°C respectively, and the throat area is 12.50 cm^2 and the exit area is 25 cm^2 estimate the exit Mach number, exit pressure, loss in stagnation pressure, and entropy increase, during the flow between the tanks.

Solution With reference to Fig. 17.18, at shock section

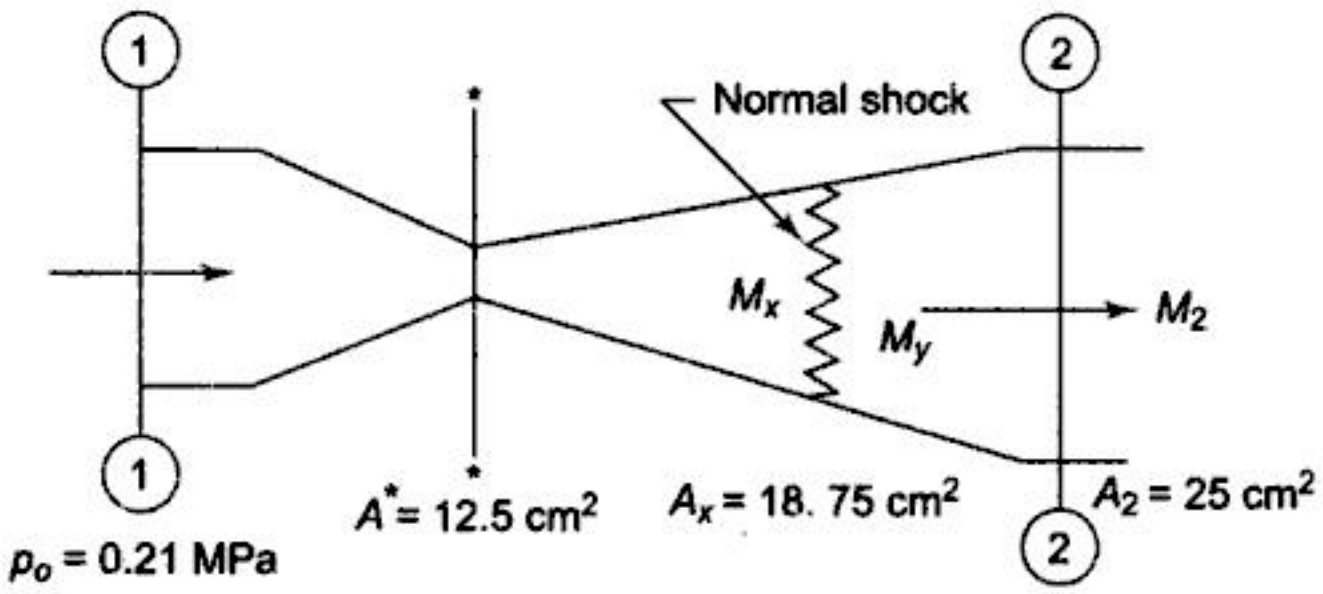

Fig. 17.18

$$\frac{A_x}{A^*} = \frac{18.75}{12.50} = 1.5$$

Upto the shock, the flow is isentropic. From the isentropic flow tables, when

$$\frac{A}{A^*} = 1.5, \qquad M_x = 1.86 \qquad \frac{p_x}{p_{ox}} = 0.159$$

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CHAPTER 18

Elements of Heat Transfer

18.1 BASIC CONCEPTS

Energy balances by first law have been made in a variety of physical situations, say, in a feedwater heater or a cooling coil. However, no indication has been given regarding the size of the heat exchanger, for heating or cooling of a fluid. If we consider a steel block heated in a furnace, to be allowed to cool in room air, we can estimate the amount of heat lost by the block in cooling by energy balance. But how long the cooling process will take place cannot be answered by thermodynamics. It is the science of heat transfer which is concerned with the estimation of the rate at which heat is transferred, the duration of heating or cooling for a certain heat duty and the surface area required to accomplish that heat duty.

There are three modes in which heat may be transferred: (a) conduction, (b) convection and (c) radiation.

(a) *Conduction* refers to the transfer of heat between two bodies or two parts of the same body through molecules which are more or less stationary. In liquids and gases conduction results from the transport of energy by molecular motion near the wall and in solids it takes place by a combination of lattice vibration and electron transport. In general, good electrical conductors are also good thermal conductors.

(b) *Convection* heat transfer occurs because of the motion of a fluid past a heated surface—the faster the motion, the greater the heat transfer. The convection heat transfer is usually assumed to be proportional to the surface area in contact with the fluid and the difference in temperature of the surface and fluid. Thus,

$$Q = hA\,[T_w - T_f]$$

where h is called the convection heat transfer coefficient, which is a strong function of both fluid properties and fluid velocity ($W/m^2\,K$).

(c) *Radiation* heat transfer is the result of electromagnetic radiation emitted by a surface because of the temperature of the surface. This differs from other forms

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or $$Q = - KA_{\text{g.m.}} \frac{t_2 - t_1}{x_w} \tag{18.7}$$

where $A_{g.m.}$ = geometrical mean area

$$= \sqrt{A_1 A_2} = \sqrt{4\pi r_1^2 \, 4\pi r_2^2} = 4\pi r_1 r_2$$

and x_w = wall thickness of the sphere = $r_2 - r_1$

Here the thermal resistance offered by the wall to heat conduction is

$$R = \frac{x_w}{KA_{\text{g.m.}}}$$

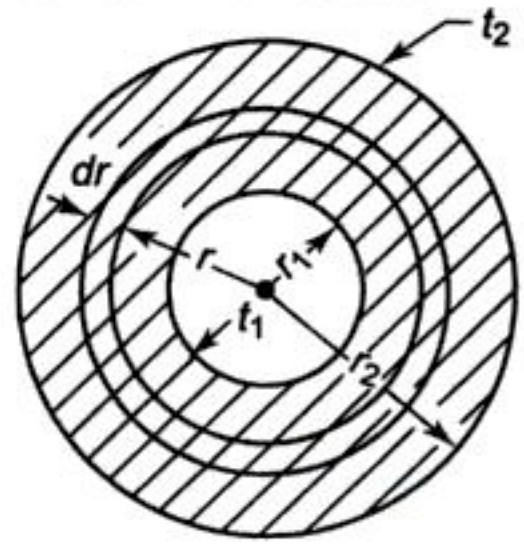

Fig. 18.8 *Heat Conduction Through a Sphere*

Thus similar expressions of thermal resistance hold good for flat plate, cylinder, and sphere, which are

$$R_{\text{plate}} = \frac{x_w}{KA}, \; R_{\text{cylinder}} = \frac{x_w}{KA_{\text{l.m.}}} \text{ and}$$

$$R_{\text{sphere}} = \frac{x_w}{KA_{\text{g.m.}}}$$

where K is the thermal conductivity of the wall material.

18.2.4 Fins

Fins are often attached with heat transfer surfaces to increase the area for convection and thus increase the heat transfer rate, particularly when heat transfer coefficient is low. One type of fin is the straight fin of a rectangular profile shown in Fig. 18.9. By combining a conduction and convection energy balance on the fin and neglecting heat loss from the tip, it is possible to derive an expression for the temperature distribution at any x position as (see "Heat Transfer" by P.K. Nag, Tata McGraw-Hill, 2002)

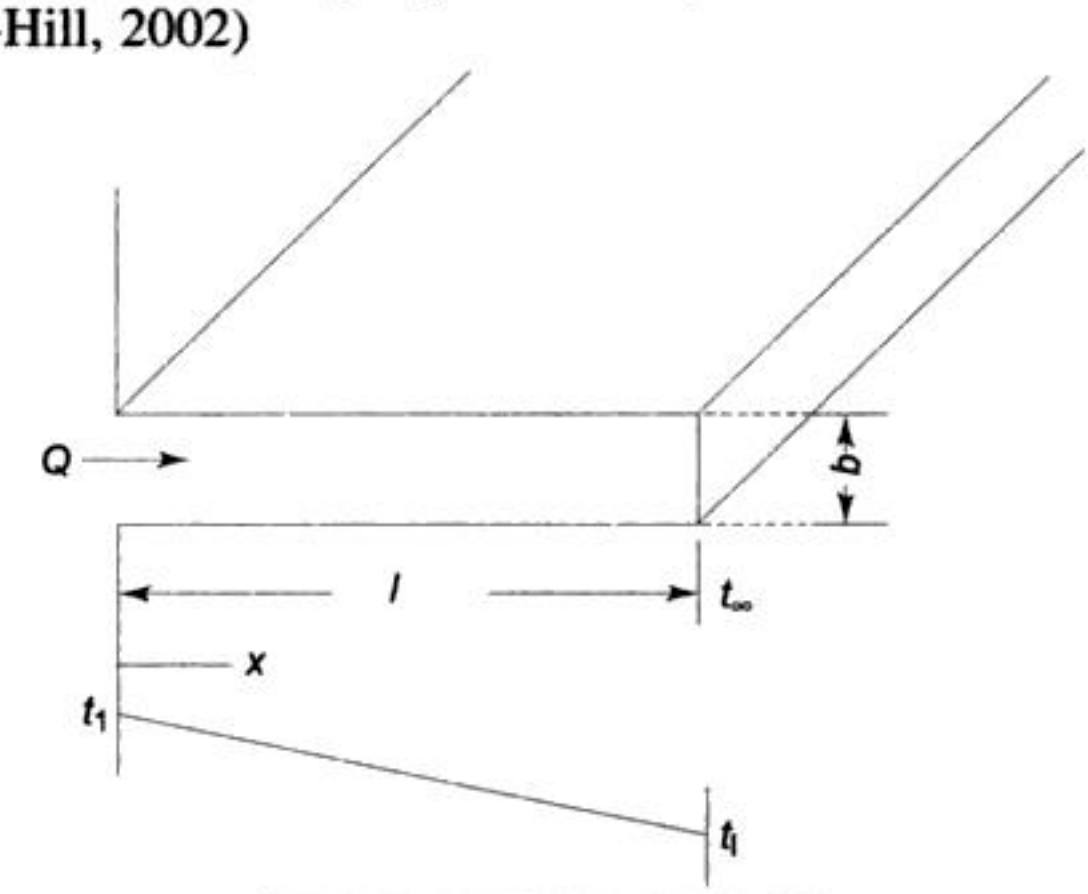

Fig. 18.9 *Rectangular Fin*

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Again

$$dQ = U_0\, dA_0\, \Delta t$$

$$-\frac{d(\Delta t)}{\mu_c} = U_0 \cdot dA_0 \cdot \Delta t$$

$$\int_{\Delta t_i}^{\Delta t_e} -\frac{d(\Delta t)}{\Delta t} = \int_i^e U_0\, dA_0 \cdot \mu_c$$

$$\ln \frac{\Delta t_i}{\Delta t_e} = U_0 \cdot A_0 \cdot \mu_c = U_0 A_0 \frac{\Delta t_i - \Delta t_e}{Q}$$

$$\therefore \qquad Q = U_0 A_0 \Delta t_{\text{l.m.}}$$

where

$$\Delta t_{\text{l.m.}} \cdot \frac{\Delta t_i - \Delta t_e}{\ln \dfrac{\Delta t_i}{\Delta t_e}}$$

$$\Delta t_i = t_{hi} - t_{c2},\ \Delta t_e = t_{h2} - t_{c1}$$

$$\frac{1}{U_0 A_0} = \frac{1}{h_i A_i} + \frac{x_w}{K_w A_{\text{l.m.}}} + \frac{1}{h_0 A_0}$$

and when x_w is small,

$$\frac{1}{U_0} \cong \frac{1}{h_i} + \frac{x_w}{K_w} + \frac{1}{h_0}$$

$$\therefore \qquad Q = U_0 A_0 \Delta t_{\text{l.m.}}$$

$$= \dot{m}_h c_h (t_{h1} - t_{h2}) = \dot{m}_c c_c (t_{c2} - t_{c1}) \qquad (18.29)$$

For the same rate of heat transfer, and inlet and exit temperatures, ($\Delta t_{\text{l.m.}}$) counterflow is greater than ($\Delta t_{\text{l.m.}}$) parallel flow. So the surface area required is less for counterflow operation. For parallel flow, $t_{h2} > t_{c2}$, i.e., the hot fluid cannot be cooled below t_{c2} or the cold fluid cannot be heated above t_{h2}. But for counterflow operation, t_{h2} may be less than t_{c2} which means that the hot fluid can be cooled below t_{c2} or the cold fluid heated above t_{h2}. For these reasons, counterflow heat exchangers are much more common in practice.

When one of the two fluids undergoes phase change (at constant temperature and pressure), e.g. condensation and evaporation, $\Delta_{\text{l.m.}}$ is the same for parallel flow the counterflow (Fig. 18.20), and the heating surface required is also the same.

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The net energy leaving the surface is:

$$\frac{Q}{A} = J - G = \varepsilon E_B + (1-\varepsilon)\, G - G$$

$$= \varepsilon(E_B - G) = \varepsilon\left[E_B - \frac{J - \varepsilon E_B}{1-\varepsilon}\right]$$

$$= \varepsilon \frac{E_B - J}{1-\varepsilon}$$

$$Q_{\text{net}} = \frac{E_B - J}{(1-\varepsilon)/A\varepsilon} \tag{18.40}$$

If the numerator is considered as the potential difference, the denominator as the "*surface resistance*" to radiation, and the heat flow as current, then a network element could be drawn as shown in Fig. 18.27.

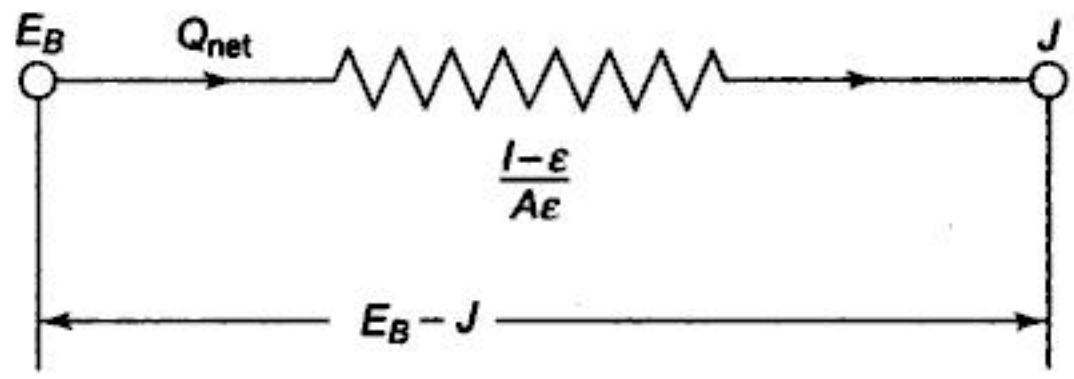

Fig. 18.27 *Surface Resistance in Radiation Network*

Let us now consider the exchange of radiant energy by two surfaces A_1 and A_2 (Fig. 18.28). Of the total radiation which leaves the surface 1, the amount that reaches surface 2 is $J_1 A_1 F_{12}$, and of the total energy leaving surface 2, the amount that reaches surface 1 is $J_2 A_2 F_{21}$.

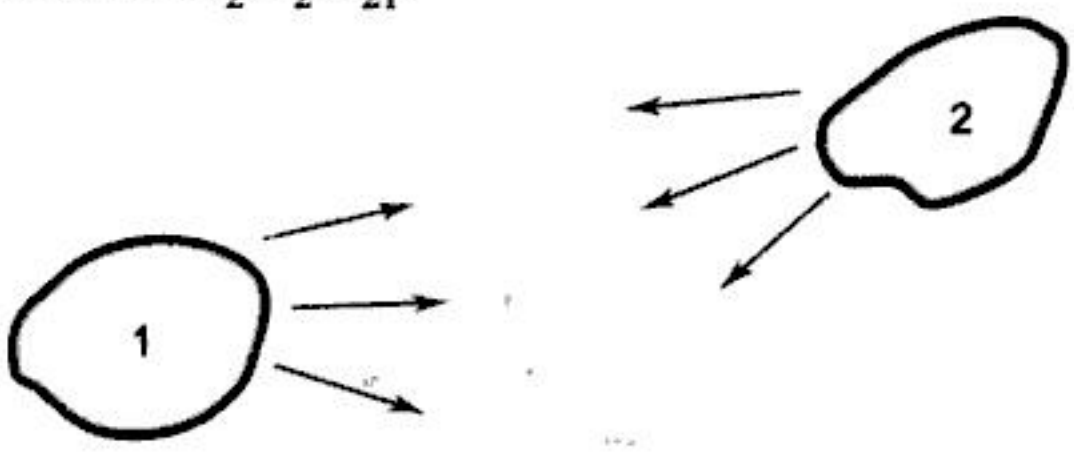

Fig. 18.28 *Radiation Interaction between Two Gray Surfaces*

The net energy interchange between two surface is:

$$Q_{1-2} = J_1 A_1 F_{12} - J_2 E_{21} A_2$$

$$= (J_1 - J_2) A_1 F_{12} = \frac{J_1 - J_2}{1/(A_1 F_{12})} \tag{18.41}$$

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$$t_{h_2} = 121 - 30.84 = \mathbf{90.16°C}$$

= exit temperature of oil.

$$Q = \dot{m}_h c_h [t_{h_1} - t_{h_2}] = 42.19 \times 30.84 = \mathbf{1308\ kW}$$

Example 18.9 Water flows inside a tube 5 cm in diameter and 3 m long at a velocity 0.8 m/s. Determine the heat transfer coefficient and the rate of heat transfer if the mean water temperature is 50°C and the wall is isothermal at 70°C. For water at 60°C, take K = 0.66 W/mK, $n = 0.478 \times 10^{-6}$ m^2/s, and Pr = 2.98.

Solution Reynolds number,

$$\text{Re} = \frac{\mu_m D}{\nu} = \frac{0.8 \times 0.05}{0.478 \times 10^{-6}} = 83{,}700$$

The flow is turbulent. Prandtl number, Pr = 2.98
From the Dittus and Boelter equation

$$\text{Nu} = 0.023\ (\text{Re})^{0.8}\ (\text{Pr})^{0.4}$$

$$\frac{hD}{K} = 0.023\ (83{,}700)^{0.8}\ (2.98)^{0.4}$$

$$\frac{h \times 0.05}{0.66} = 0.023 \times 8673.6 \times 1.5478$$

$\therefore$ $h = \mathbf{4075\ W/m^2\ K}$

$\therefore$ $Q = hA\ (t_w - t_f) = 4075 \times \pi \times 0.05 \times 3 \times 20$

$= 38387\ \text{W} = \mathbf{38.39\ kW}$

Example 18.10 An electrically heated plate 15 cm high and 10 cm wide is maintained at 140°C. Estimate the rate of heat dissipation from both sides of the plate in an atmosphere at 20°C, if the radiation heat transfer coefficient is 8.72 W/m^2 K. For air at the mean temperature of 80°C, take $\nu = 21.09 \times 10^{-6}$ m^2/s, Pr = 0.692, and K = 0.0305 W/mK.

Solution

$$Q = (h_c + h_r)\, A(t_w - t_f)$$

where $A = 0.1 \times 0.15 \times 2 = 0.3\ \text{m}^2$

$h_r = 8.72\ \text{W/m}^2\ \text{K}$, $t_w - t_f = 140 - 20 = 120°\text{C}$

$$\beta = \frac{1}{\text{v}}\left(\frac{\partial \text{v}}{\partial \text{T}}\right)_p = \frac{1}{T_{\text{mean}}} = \frac{1}{273 + 80} = \frac{1}{353}\ (K)^{-1}$$ $L = 0.15$ m

$\therefore$ Grashoff number, $\text{Gr} = \dfrac{g\beta\, \Delta\, TL^3}{\nu^2}$

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CHAPTER 19

Gas Compressors

A gas compressor is a device in which work is done in the gas to raise its pressure, with an appreciable increase in its density. The compression of gases is an important process in many power plants, refrigeration plants, and industrial plants. Industrial uses occur in connection with compressed air motors for tools, air brakes for vehicles, servo-mechanisms, metallurgical and chemical processes, conveying of materials through ducts, transporting of natural gas, and production of bottled gases.

Compressors can be (a) positive displacement machines like reciprocating compressors, Root's blower and vane-sealed machines and (b) turbine type like centrifugal and axial flow compressors.

19.1 COMPRESSION PROCESSES

A gas compression process may be either adiabatic or involving heat transfer, depending upon the purpose for which the gas is compressed. If the compressed gas is to be used promptly in an engine or in a combustion process, adiabatic compression is desirable so as to obtain the maximum energy in the gas at the end of compression. In many applications, however, the gas is not used promptly but is stored in a tank or receiver for use later when needed. The gas in the tank transfer heat to the surroundings so that when finally used it is at room temperature. In such cases the objective of the compression and storage process is simply to increase the pressure of the gas without change of temperature. It will be shown below that if the gas is cooled during compression, the work required will be less than for adiabatic compression. Another advantage of cooling is the reduction in volume of gas and less pipe friction losses. Since cooling during the compression process is not very effective, *after-coolers* are often used to cool the gas leaving the compressor.

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driven by a constant speed motor. The compressor cylinder of Fig. 19.7 is fitted with a clearance pocket which can be opened at will by a valve. Let us suppose that this machine is operating at conditions corresponding to line *e* in Fig. 19.6. If the clearance volume is at minimum value *a* the volumetric efficiency and the flow through the machine are maximum. If the clearance pocket is then opened to increase the clearance to *b*, the volumetric efficiency and the flow are reduced. By increasing the clearance in steps, as indicated by points *c* and *d*, the flow may be reduced in steps to zero. The work per kg of gas compressed is, however, not affected by the clearance volume in an idealized compressor.

For a given pressure ratio, η_{vol} is zero when the maximum clearance is

$$C_{max} = \frac{1}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1} \tag{19.14}$$

19.4.1 Effect of Pressure Ratio on Volumetric Efficiency

It is evident from Fig. 19.6 that as the pressure ratio is increased the volumetric efficiency of a compressor of fixed clearance decreases, eventually becoming zero. This can also be seen in an indicator diagram, Fig. 19.8. As the discharge pressure is increased from p_{2a} to p_{2b}, the volume V_{1a} taken at pressure p_1 decreases to V_{1b}, and η_{vol} decreases. At some pressure p_{2c} the compression line intersects the line of clearance volume and there is no discharge of gas. An attempt to pump to p_{2c} (or any higher pressure) would result in compression and re-expansion of the same gas repeatedly, with no flow in or out.

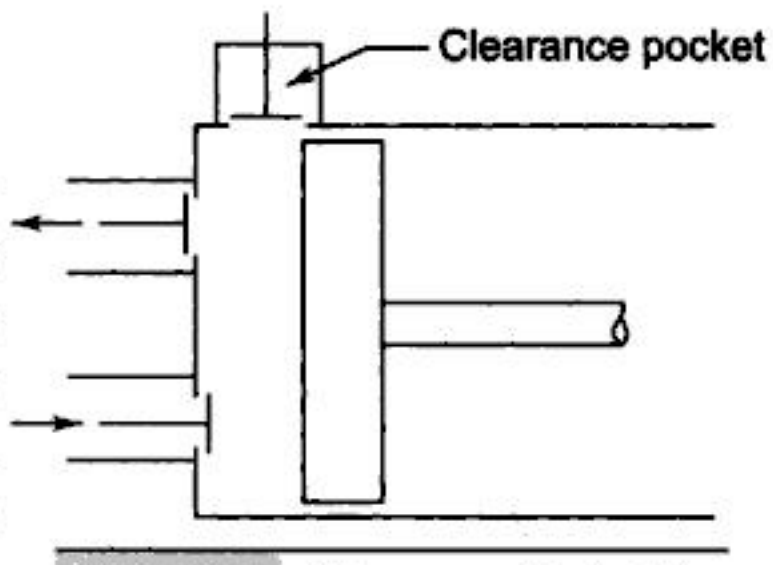

Fig. 19.7 *Clearance Pocket for Capacity Control*

The maximum pressure ratio attainable with a reciprocating compressor cylinder is then seen to be limited by the clearance, which is given from Eq. (19.3), as

$$\frac{p_{2max}}{p_1} = \left(1 + \frac{1}{C}\right)^n \tag{19.15}$$

19.5 MULTI-STAGE COMPRESSION

When compressing a gas (or air) to high pressure it is advantageous to do it in stages. The condition for minimum work requires the compression to be isothermal. Since the temperature after compression is given by $T_2 = T_1 (p_2/p_1)^{n-1/n}$, the delivery temperature,

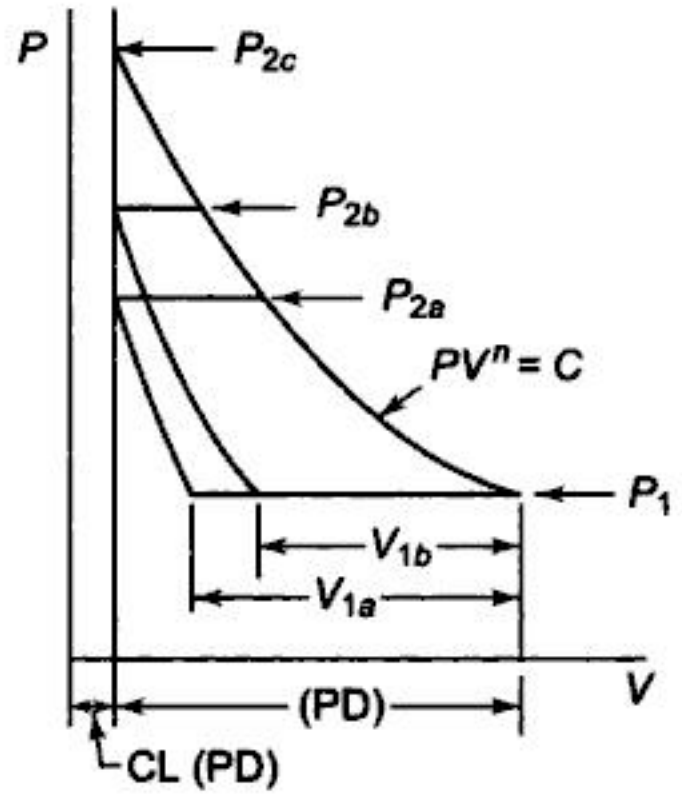

Fig. 19.8 *Effect of Pressure-Ratio on Capacity*

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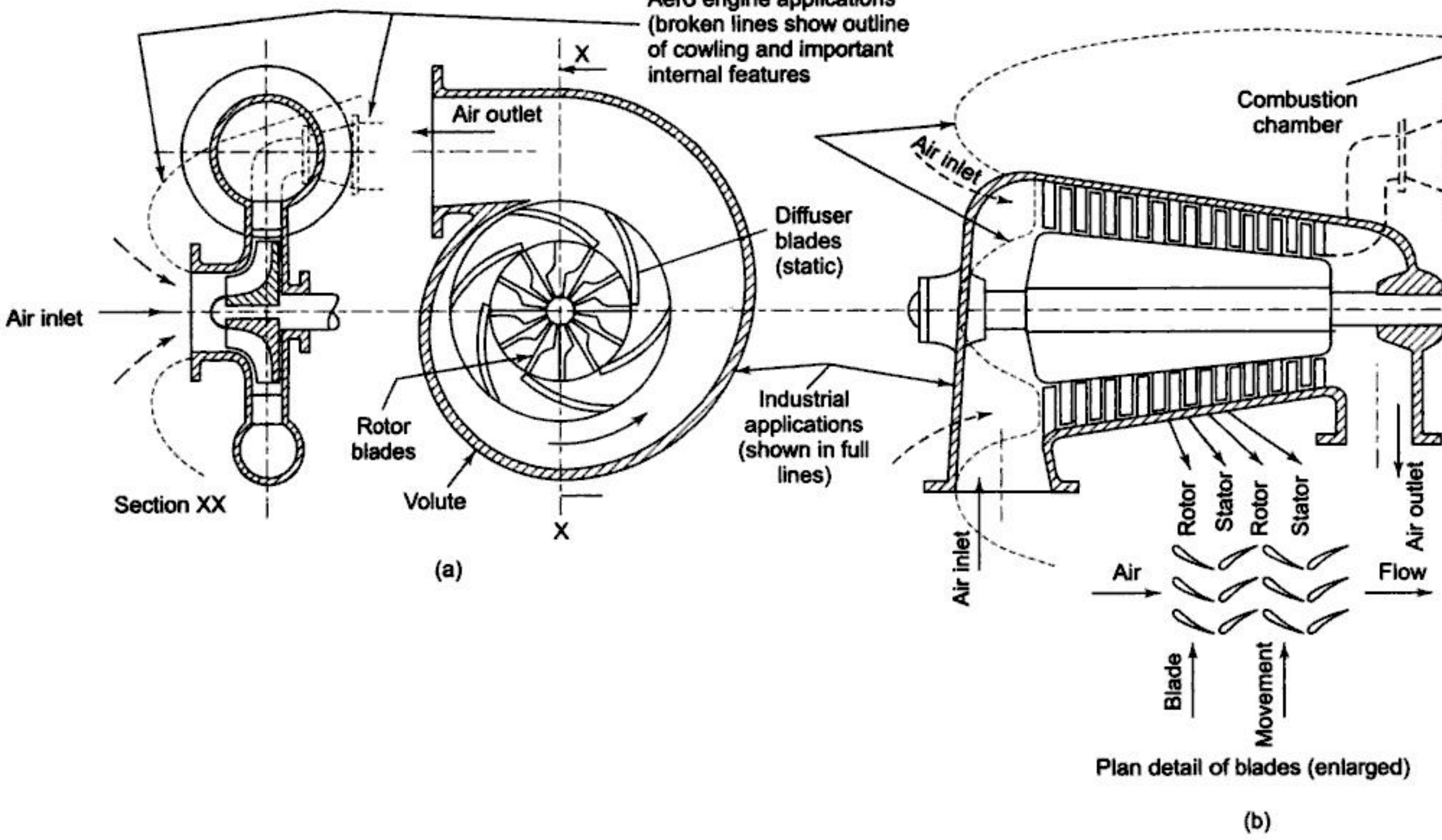

Fig. 19.16 *(a) Centrifugal Compressor, (b) Axial-Flow Compressor*

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Solution

$$\dot{V} = \frac{\dot{m}RT_1}{p_1} = \frac{1 \times 0.29 \times 343}{100} = 0.995 \text{ m}^3/\text{s}$$

∴ Power required by the Roots blower

$$= \dot{V}\ (p_2 - p_1) = 0.995 \times 100 = \mathbf{99.5\ kW}$$

For the vane compressor,

$$p_1\ V_1^{\gamma} = p_2\ V_2^{\gamma}$$

$$\therefore \qquad p_2 = p_1 \left(\frac{V_1}{V_2}\right)^{\gamma} = 1 \times \left(\frac{1}{0.7}\right)^{1.4} = 1.65 \text{ bar}$$

$$\dot{V}_2 = 0.7\ \dot{V}_1 = 0.7 \times 0.995 = 0.696 \text{ m}^3/\text{s}$$

$$\text{Power required} = \frac{\gamma}{\gamma - 1} p_1 \dot{V}_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + \dot{V}_2\ (p_3 - p_2)$$

$$= \frac{1.4}{0.4} \times 100 \times 0.995 \left[\left(\frac{1.65}{1}\right)^{\frac{0.4}{1.4}} - 1\right]$$

$$+ 0.696\ (2 - 1.65) \times 100 = \mathbf{78\ kW}$$

Example 19.9 A gas turbine utilizes a two-stage centrifugal compressor. The pressure ratios for the first and second stages are 2.5 to 1 and 2.1 to 1 respectively. The flow of air is 5 kg/s, this air being drawn at 1.013 bar, and 10°C. If the temperature drop in the intercooler is 50°C and the isentropic efficiency is 85% for each stage, calculate (a) the actual temperatures at the end of each stage, (b) the total compressor power. Take $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg K.

Solution

Stage I

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{(Fig. 19.23)}$$

$$T_{2s} = 283\ (2.5)^{0.4/1.4} = 367.7 \text{ K}$$

$$T_2 - T_3 = 50°\text{C}$$

$$T_{2s} - T_1 = \eta_s\ (T_2 - T_1)$$

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_s}$$

$$= 283 + \frac{367.7 - 283}{0.85} = 382.65 \text{ K}$$

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Appendix A

A.1 Steam Tables*

Table A.1.1 *Saturated Steam: Temperature Table*

Temp. °C T	Specific Volume, m^3/kg			Internal Energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/kg K		
	Pressure kPa, MPa P	Sat. Liquid v_f	Sat. Vapour v_g	Sat. Liquid u_f	Evap. u_{fg}	Sat. Vapour u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapour h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat Vapour s_g
0.01	0.6113	0.001000	206.132	0.00	2375.3	2375.3	0.00	2501.3	2501.3	0.0000	9.1562	9.1562
5	0.8721	0.001000	147.118	20.97	2361.3	2382.2	20.98	2489.6	2510.5	0.0761	8.9496	9.0257
10	1.2276	0.001000	106.377	41.99	2347.2	2389.2	41.99	2477.7	2519.7	0.1510	8.7498	8.9007
15	1.7051	0.001001	77.925	62.98	2333.1	2396.0	62.98	2465.9	2528.9	0.2245	8.5569	8.7813
20	2.3385	0.001002	57.790	83.94	2319.0	2402.9	83.94	2454.1	2538.1	0.2966	8.3706	8.6671
25	3.1691	0.001003	43.359	104.86	2304.9	2409.8	104.87	2442.3	2547.2	0.3673	8.1905	8.5579
30	4.2461	0.001004	32.893	125.77	2290.8	2416.6	125.77	2430.5	2556.2	0.4369	8.0164	8.4533
35	5.6280	0.001006	25.216	146.65	2276.7	2423.4	146.66	2418.6	2565.3	0.5052	7.8478	8.3530
40	7.3837	0.001008	19.523	167.53	2262.6	2430.1	167.54	2406.7	2574.3	0.5724	7.6845	8.2569
45	9.5934	0.001010	15.258	188.41	2248.4	2436.8	188.42	2394.8	2583.2	0.6386	7.5261	8.1647
50	12.350	0.001012	12.032	209.30	2234.2	2443.5	209.31	2382.7	2592.1	0.7037	7.3725	8.0762
55	15.758	0.001015	9.568	230.19	2219.9	2450.1	230.20	2370.7	2600.9	0.7679	7.2234	7.9912
60	19.941	0.001017	7.671	251.09	2205.5	2456.6	251.11	2358.5	2609.6	0.8311	7.0784	7.9095
65	25.033	0.001020	6.197	272.00	2191.1	2463.1	272.03	2346.2	2618.2	0.8934	6.9375	7.8309
70	31.188	0.001023	5.042	292.93	2176.6	2469.5	292.96	2333.8	2626.8	0.9548	6.8004	7.7552
75	38.578	0.001026	4.131	313.87	2162.0	2475.9	313.91	2321.4	2635.3	1.0154	6.6670	7.6824
80	47.390	0.001029	3.407	334.84	2147.4	2482.2	334.88	2308.8	2643.7	1.0752	6.5369	7.6121
85	57.834	0.001032	2.828	355.82	2132.6	2488.4	355.88	2296.0	2651.9	1.1342	6.4102	7.5444
90	70.139	0.001036	2.361	376.82	2117.7	2494.5	376.90	2283.2	2660.1	1.1924	6.2866	7.4790
95	84.554	0.001040	1.982	397.86	2102.7	2500.6	397.94	2270.2	2668.1	1.2500	6.1659	7.4158
100	0.10135	0.001044	1.6729	418.91	2087.6	2506.5	419.02	2257.0	2676.0	1.3068	6.0480	7.3548

* Adapted from Joseph H. Keenan, Frederick G.Keyes, Philip G. Hill, and Joan G. Moore, *Steam Tables*, John Wiley and Sons, New York, 1969.

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Table A.1.2 *(Contd.)*

Pressure	*Temp.*	*Specific Volume, m^3/kg*		*Internal Energy, kJ/kg*			*Enthalpy, kJ/kg*			*Entropy, kJ/kg K*		
		Sat.	*Sat.*	*Sat.*		*Sat.*	*Sat.*		*Sat.*	*Sat.*		*Sat*
MPa	*°C*	*Liquid*	*Vapour*	*Liquid*	*Evap.*	*Vapour*	*Liquid*	*Evap.*	*Vapour*	*Liquid*	*Evap.*	*Vapour*
P	*T*	v_f	v_g	u_f	u_{fg}	u_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
2.50	223.99	0.001197	0.07998	959.09	1644.0	2603.1	962.09	1841.0	2803.1	2.5546	3.7028	6.2574
2.75	229.12	0.001207	0.07275	982.65	1621.2	2603.8	985.97	1817.9	2803.9	2.6018	3.6190	6.2208
3.00	233.90	0.001216	0.06668	1004.76	1599.3	2604.1	1008.41	1795.7	2804.1	2.6456	3.5412	6.1869
3.25	238.38	0.001226	0.06152	1025.62	1578.4	2604.0	1029.60	1774.4	2804.0	2.6866	3.4685	6.1551
3.50	242.60	0.001235	0.05707	1045.41	1558.3	2603.7	1049.73	1753.7	2803.4	2.7252	3.4000	6.1252
4.0	250.40	0.001252	0.049778	1082.28	1520.0	2602.3	1087.29	1714.1	2801.4	2.7963	3.2737	6.0700
5.0	263.99	0.001286	0.039441	1147.78	1449.3	2597.1	1154.21	1640.1	2794.3	2.9201	3.0532	5.9733
6.0	275.64	0.001319	0.032440	1205.41	1384.3	2589.7	1213.32	1571.0	2784.3	3.0266	2.8625	5.8891
7.0	285.88	0.001351	0.027370	1257.51	1323.0	2580.5	1266.97	1505.1	2772.1	3.1210	2.6922	5.8132
8.0	295.06	0.001384	0.023518	1305.54	1264.3	2569.8	1316.61	1441.3	2757.9	3.2067	2.5365	5.7431
9.0	303.40	0.001418	0.020484	1350.47	1207.3	2557.8	1363.23	1378.9	2742.1	3.2857	2.3915	5.6771
10.0	311.06	0.001452	0.018026	1393.00	1151.4	2544.4	1407.53	1317.1	2724.7	3.3595	2.2545	5.6140
11.0	318.15	0.001489	0.015987	1433.68	1096.1	2529.7	1450.05	1255.5	2705.6	3.4294	2.1233	5.5527
12.0	324.75	0.001527	0.014263	1472.92	1040.8	2513.7	1491.24	1193.6	2684.8	3.4961	1.9962	5.4923
13.0	330.93	0.001567	0.012780	1511.09	985.0	2496.1	1531.46	1130.8	2662.2	3.5604	1.8718	5.4323
14.0	336.75	0.001611	0.011485	1548.53	928.2	2476.8	1571.08	1066.5	2637.5	3.6231	1.7485	5.3716
15.0	342.24	0.001658	0.010338	1585.58	869.8	2455.4	1610.45	1000.0	2610.5	3.6847	1.6250	5.3097
16.0	347.43	0.001711	0.009306	1622.63	809.1	2431.7	1650.00	930.6	2580.6	3.4760	1.4995	5.2454
17.0	352.37	0.001770	0.008365	1660.16	744.8	2405.0	1690.25	856.9	2547.2	3.8078	1.3698	5.1776
18.0	357.06	0.001840	0.007490	1698.86	675.4	2374.3	1731.97	777.1	2509.1	3.8713	1.2330	5.1044
19.0	361.54	0.001924	0.006557	1739.87	598.2	2338.1	1776.43	688.1	2464.5	3.9387	1.0841	5.0227
20.0	365.81	0.002035	0.005834	1785.47	507.6	2293.1	1826.18	583.6	2409.7	4.0137	0.9132	4.9269
21.0	369.89	0.002206	0.004953	1841.97	388.7	2230.7	1888.30	446.4	2334.7	4.1073	0.6942	4.8015
22.0	373.80	0.002808	0.003526	1973.16	108.2	2081.4	2034.92	124.0	2159.0	4.3307	0.1917	4.5224
22.09	374.14	0.003155	0.003155	2029.58	0	2029.6	2099.26	0	2099.3	4.4297	0	4.4297

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Table A.1.3 *(Contd.)*

T	P = 9.00 MPa (303.40)				P = 10.00 MPa (311.06)				P = 12.50 MPa (327.89)			
	v	u	h	s	v	u	h	s	v	u	h	s
800	0.05409	3632.5	4199.4	7.4597	0.04859	3629.0	4114.9	7.4077	0.03869	3620.0	4103.7	7.2965
900	0.5950	3829.2	4364.7	7.6782	0.05349	3826.3	4361.2	7.6272	0.04267	3819.1	4352.5	7.5181
1000	0.06485	4030.3	4613.9	7.8821	0.05832	4027.8	4611.0	7.8315	0.04658	4021.6	4603.8	7.7237
1100	0.07016	4236.3	4867.7	8.0739	0.06312	4234.0	4865.1	8.0236	0.05045	4228.2	4858.8	7.9165
1200	0.07544	4447.2	5126.2	8.2556	0.06789	4444.9	5123.8	8.2054	0.05430	4439.3	5118.0	8.0987
1300	0.08072	4662.7	5389.2	8.4283	0.07265	4660.4	5387.0	8.3783	0.05813	4654.8	5381.4	8.2717
	P = 15 MPa (342.24)				P = 17.5 MPa (354.75)				P = 20 MPa (365.81)			
Sat.	.010338	2455.4	2610.5	5.3097	.0079204	2390.2	2528.8	5.1418	.0058342	2293.1	2409.7	4.9269
350	.011470	2520.4	2692.4	5.4420	.0017139	1632.0	1662.0	3.7612	.0016640	1612.3	1645.6	3.7280
400	.015649	2740.7	2975.4	5.8810	.0124477	2685.0	2902.8	5.7212	.0099423	2619.2	2818.1	5.5539
450	.018446	2879.5	3156.2	6.1403	.0151740	2844.2	3109.7	6.0182	.0126953	2806.2	3060.1	5.9016
500	.020800	2996.5	3308.5	6.3442	.0173585	2970.3	3274.0	6.2382	.0147683	2942.8	3238.2	6.1400
550	.022927	3104.7	3448.6	6.5198	.0192877	3083.8	3421.4	6.4229	.0165553	3062.3	3393.5	6.3347
600	.024911	3208.6	3582.3	6.6775	.0210640	3191.5	3560.1	6.5866	.0181781	3174.0	3537.6	6.5048
650	.026797	3310.4	3712.3	6.8223	.0227372	3296.0	3693.9	6.7356	.0196929	3281.5	3675.3	6.6582
700	.028612	3410.9	3840.1	6.9572	.0243365	3398.8	3824.7	6.8736	.0211311	3386.5	3809.1	6.7993
800	.032096	3611.0	4092.4	7.2040	.0273849	3601.9	4081.1	7.1245	.0238532	3592.7	4069.8	7.0544
900	.035457	3811.9	4343.8	7.4279	.0303071	3804.7	4335.1	7.3507	.0264463	3797.4	4326.4	7.2830
1000	.038748	4015.4	4596.6	7.6347	.0331580	4009.3	4589.5	7.5588	.0289666	4003.1	4582.5	7.4925
1100	.042001	4222.6	4852.6	7.8282	.0359695	4216.9	4846.4	7.7530	.0314471	4211.3	4840.2	7.6874
1200	.045233	4433.8	5112.3	8.0108	.0387605	4428.3	5106.6	7.9359	.0339071	4422.8	5101.0	7.8706
1300	.048455	4649.1	5375.9	8.1839	.0415417	4643.5	5370.5	8.1093	.0363574	4638.0	5365.1	8.0441

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Table A.2.1 *(Contd.)*

T	P	v_f	v_g	h_f	h_{fg}	h_g	s_f	s_g
– 35	0.0807	0.666	0.195398	4.416	167.368	171.784	0.0187	0.7214
– 30	0.1004	0.672	0.159375	8.854	165.222	174.076	0.0371	0.7165
– 25	0.1237	0.679	0.131166	13.315	163.037	176.352	0.0552	0.7121
– 20	0.1509	0.685	0.108847	17.800	160.810	178.610	0.0730	0.7082
– 15	0.1826	0.693	0.091018	22.312	158.534	180.846	0.0906	0.7046
– 10	0.2191	0.700	0.076646	26.851	156.207	183.058	0.1079	0.7014
– 5	0.2610	0.708	0.064963	31.420	153.823	185.243	0.1250	0.6986
0	0.3086	0.716	0.055389	36.022	151.376	187.397	0.1418	0.6960
5	0.3626	0.724	0.047485	40.659	148.859	189.518	0.1585	0.6937
10	0.4233	0.733	0.040914	45.337	146.265	191.602	0.1750	0.6916
15	0.4914	0.743	0.035413	50.058	143.586	193.644	0.1914	0.6897
20	0.5673	0.752	0.030780	54.828	140.812	195.641	0.2076	0.6879
25	0.6516	0.763	0.026854	59.653	137.933	197.586	0.2237	0.6863
30	0.7449	0.774	0.023508	64.539	134.936	199.475	0.2397	0.6848
35	0.8477	0.786	0.020641	69.494	131.805	201.299	0.2557	0.6834
40	0.9607	0.798	0.018171	74.527	128.525	203.051	0.2716	0.6820
45	1.0843	0.811	0.016032	79.647	125.074	204.722	0.2875	0.6806
50	1.2193	0.826	0.014170	84.868	121.430	206.298	0.3034	0.6792
55	1.3663	0.841	0.012542	90.201	117.565	207.766	0.3194	0.6777
60	1.5259	0.858	0.01111	95.665	113.443	209.109	0.3355	0.6760
65	1.6988	0.877	0.009847	101.279	109.024	210.303	0.3518	0.6742
70	1.8858	0.897	0.008725	107.067	104.255	211.321	0.3683	0.6721
75	2.0874	0.920	0.007723	113.058	99.068	212.126	0.3851	0.6697
80	2.3046	0.946	0.006821	119.291	93.373	212.665	0.4023	0.6667
85	2.5380	0.976	0.006005	125.818	87.047	212.865	0.4201	0.6631
90	2.7885	1.012	0.005258	132.708	79.907	212.614	0.4385	0.6585
95	3.0569	1.056	0.004563	140.068	71.658	211.726	0.4579	0.6526
100	3.3440	1.113	0.003903	148.076	61.768	209.843	0.4788	0.6444
105	3.6509	1.197	0.003242	157.085	49.014	206.099	0.5023	0.6319
110	3.9784	1.364	0.002462	168.059	28.425	196.484	0.5322	0.6064

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Table A.3.2 *Superheated Refrigerant-22*

Temp. °C	*v* m^3/kg	*h* kJ/kg	*s.* kJ/kg K	*v* m^3/kg	*h* kJ/kg	*s.* kJ/kg K	*v* m^3/kg	*h* kJ/kg	*s* kJ/kg K
	0.05 MPa			*0.10 MPa*			*0.15 MPa*		
– 40	0.440633	234.724	1.07616	0.216331	233.337	1.00523	—	—	—
– 30	0.460641	240.602	1.10084	0.226754	239.359	1.03052	0.148723	238.078	0.98773
– 20	0.480543	246.586	1.12495	0.237064	245.466	1.05513	0.155851	244.319	1.01288
– 10	0.500357	252.676	1.14855	0.247279	251.665	1.07914	0.162879	250.631	1.03733
0	0.520095	258.874	1.17166	0.257415	257.956	1.10261	0.169823	257.022	1.06116
10	0.539771	265.180	1.19433	0.267485	264.345	1.12558	0.176699	263.496	1.08444
20	0.559393	271.594	1.21659	0.277500	270.831	1.14809	0.183516	270.057	1.10721
30	0.578970	278.115	1.23846	0.287467	277.416	1.17017	0.190284	276.709	1.12952
40	0.598507	284.743	1.25998	0.297394	284.101	1.19187	0.197011	283.452	1.15140
50	0.618011	291.478	1.28114	0.307287	290.887	1.21320	0.203702	290.289	1.17289
60	0.637485	298.319	1.30199	0.317149	297.772	1.23418	0.210362	297.220	1.19402
70	0.656935	305.265	1.32253	0.326986	304.757	1.25484	0.216997	304.246	1.21479
80	0.676362	312.314	1.34278	0.336801	311.842	1.27519	0.223608	311.368	1.23525
90	0.695771	319.465	1.36275	0.346596	319.026	1.29524	0.230200	318.584	1.25540
	0.20 MPa			*0.25 MPa*			*0.30 MPa*		
– 20	0.115203	243.140	0.98184	—	—	—	—	—	—
– 10	0.120647	249.574	1.00676	0.95280	248.492	0.98231	0.078344	247.382	0.96170
0	0.126003	256.069	1.03098	0.099689	255.097	1.00695	0.082128	254.104	0.98677
10	0.131286	262.633	1.05458	0.104022	261.755	1.03089	0.085832	260.861	1.01106

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A.4 Thermodynamic Properties of Refrigerant-134a (1,1,1,2–tetrafluoroethane)

Table A.4.1 *Saturated R-134a*

		Specific Volume, m^3/kg			*Enthalpy, kJ/kg*			*Entropy, kJ/kg K*		
Temp. °C	*Abs. Press. MPa P*	*Sat. Liquid v_f*	*Evap. v_{fg}*	*Sat. Vapour v_g*	*Sat. Liquid h_f*	*Evap. h_{fg}*	*Sat. Vapour h_g*	*Sat. Liquid s_f*	*Evap. s_{fg}*	*Sat. Vapour s_g*
– 33	0.0737	0.000718	0.25574	0.25646	157.417	220.491	377.908	0.8346	0.9181	1.7528
– 30	0.0851	0.000722	0.22330	0.22402	161.118	218.683	379.802	0.8499	0.8994	1.7493
26.25	0.1013	0.000728	0.18947	0.19020	165.802	216.360	382.162	0.8690	0.8763	1.7453
– 25	0.0173	0.000730	0.17956	0.18029	167.381	215.569	382.950	0.8754	0.8687	1.7441
– 20	0.1337	0.000738	0.14575	0.14649	173.744	212.340	386.083	0.9007	0.8388	1.7395
– 15	0.1650	0.000746	0.11932	0.12007	180.193	209.004	389.197	0.9258	0.8096	1.7354
– 10	0.2017	0.000755	0.098454	0.099209	186.721	205.564	392.285	0.9507	0.7812	1.7319
– 5	0.2445	0.000764	0.081812	0.082576	193.324	202.016	395.340	0.9755	0.7534	1.7288
0	0.2940	0.000773	0.068420	0.069193	200.000	198.356	398.356	1.0000	0.7262	1.7262
5	0.3509	0.000783	0.057551	0.058334	206.751	194.572	401.323	1.0243	0.6995	1.7239
10	0.4158	0.000794	0.048658	0.049451	213.580	190.652	404.233	1.0485	0.6733	1.7218
15	0.4895	0.000805	0.041326	0.042131	220.492	186.582	407.075	1.0725	0.6475	1.7200
20	0.5728	0.000817	0.035238	0.036055	227.493	182.345	409.838	1.0963	0.6220	1.7183
25	0.6663	0.000829	0.030148	0.030977	234.590	177.920	412.509	1.1201	0.5967	1.7168
30	0.7710	0.000843	0.025865	0.026707	241.790	173.285	415.075	1.1437	0.5716	1.7153
35	0.8876	0.000857	0.022237	0.023094	249.103	168.415	417.518	1.1673	0.5465	1.7139
40	1.0171	0.000873	0.019147	0.020020	256.539	163.282	419.821	1.1909	0.5214	1.7123
45	1.1602	0.000890	0.016499	0.017389	264.110	157.852	421.962	1.2145	0.4962	1.7106
50	1.3180	0.000908	0.014217	0.015124	271.830	152.085	423.915	1.2381	0.4706	1.7088
55	1.4915	0.000928	0.012237	0.013166	279.718	145.933	425.650	1.2619	0.4447	1.7066
60	1.6818	0.000951	0.010511	0.011462	287.794	139.336	427.130	1.2857	0.4182	1.7040

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Table A.4.2 (Contd.)

Temp. °C	*v* m^3/kg	*h* *kJ/kg*	*s* *kJ/kg K*	*v* m^3/kg	*h* *kJ/kg*	*s* *kJ/kg K*	*v* m^3/kg	*h* *kJ/kg*	*s* *kJ/kg K*
	0.50 MPa			*0.60 MPa*			*0.70 MPa*		
20	0.042256	411.645	1.73420	—	—	—	—	—	—
30	0.044457	421.221	1.76632	0.036094	419.093	1.74610	0.030069	416.809	1.72770
40	0.046557	430.720	1.79715	0.037958	428.881	1.77786	0.031781	426.933	1.76056
50	0.048581	440.205	1.82696	0.039735	438.589	1.80838	0.033392	436.895	1.79187
60	0.050547	449.718	1.85596	0.041447	448.279	1.83791	0.034929	446.782	1.82201
70	0.052467	459.290	1.88426	0.043108	457.994	1.86664	0.036410	456.655	1.85121
80	0.054351	468.942	1.91199	0.044730	467.764	1.89471	0.037848	466.554	1.87964
90	0.056205	478.690	1.93921	0.046319	477.611	1.92220	0.039251	476.507	1.90743
100	0.058035	488.546	1.96598	0.047883	487.550	1.94920	0.040627	486.535	1.93467
110	0.059845	498.518	1.99235	0.049426	497.594	1.97576	0.041980	496.654	1.96143
120	0.061639	508.613	2.01836	0.050951	507.750	2.00193	0.043314	506.875	1.98777
130	0.063418	518.835	2.04403	0.052461	518.026	2.02774	0.044633	517.207	2.01372
140	0.065184	529.187	2.06940	0.053958	528.425	2.05322	0.045938	527.656	2.03932
	0.80 MPa			*0.90 MPa*			*1.00 MPa*		
40	0.027113	424.860	1.74457	0.023446	422.642	1.72943	0.020473	420.249	1.71479
50	0.028611	435.114	1.77680	0.024868	433.235	1.76273	0.021849	431.243	1.74936
60	0.030024	445.223	1.80761	0.026192	443.595	1.79431	0.023110	441.890	1.78181
70	0.031375	455.270	1.83732	0.027447	453.835	1.82459	0.024293	452.345	1.81273
80	0.032678	465.308	1.86616	0.028649	464.025	1.85387	0.025417	462.703	1.84248
90	0.033944	475.375	1.89427	0.029810	474.216	1.88232	0.026497	473.027	1.87131
100	0.035180	485.499	1.92177	0.030940	484.441	1.91010	0.027543	483.361	1.89938
110	0.036392	495.698	1.94874	0.032043	494.726	1.93730	0.028561	493.736	1.92682
120	0.037584	505.988	1.97525	0.033126	505.088	1.96399	0.029556	504.175	1.95371
130	0.038760	516.379	2.00135	0.034190	515.542	1.99025	0.030533	514.694	1.98013
140	0.039921	526.880	2.02708	0.035241	526.096	2.01611	0.031495	525.305	2.00613
150	0.041071	537.496	2.05247	0.036278	536.760	2.04161	0.032444	536.017	2.03175

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Table A.5.1 *(Contd.)*

T (°C)	*P* (kPa)	*Specific volume (m^3/kg)*			*Enthalpy (kJ/kg)*			*Entropy (kJ/kg K)*	
		Sat. liquid vt	*t*	*Sat. vapour, vg*	*Sat. liquid, ht*	*Evap., htg*	*Sat. vapour, hg*	*Sat. liquid, st*	*Sat. vapour sg*
– 16	226.45	0.001515		0.5296	107.8	1316.5	1424.4	0.4393	5.5600
– 14	246.51	0.001521		0.4889	116.9	1310.0	1427.0	0.4744	5.5305
– 12	267.95	0.001528		0.4520	126.0	1303.5	1429.5	0.5093	5.5015
– 10	290.95	0.001534		0.4185	135.2	1296.8	1432.0	0.5440	5.4730
– 8	315.25	0.001540		0.3878	144.3	1290.1	1434.4	0.5785	5.4449
– 6	341.25	0.001546		0.3599	153.5	1283.3	1436.8	0.6128	5.4173
– 4	368.90	0.001553		0.3343	162.7	1276.4	1439.1	0.6469	5.3901
– 2	398.27	0.001559		0.3109	171.9	1269.4	1441.3	0.6808	5.3633
0	429.44	0.001566		0.2895	181.1	1262.4	1443.5	0.7145	5.3369
2	462.49	0.001573		0.2698	190.4	1255.2	1445.6	0.7481	5.3108
4	497.49	0.001580		0.2517	199.6	1243.0	1447.6	0.7815	5.2852
6	531.51	0.001587		0.2351	208.9	1240.6	1449.6	0.8148	5.2599
8	573.64	0.001594		0.2198	218.3	1233.2	1451.5	0.8479	5.2350
10	614.95	0.001601		0.2056	227.6	1225.7	1453.3	0.8808	5.2104
12	658.52	0.001608		0.1926	237.0	1218.1	1455.1	0.9136	5.1861
14	704.44	0.001616		0.1805	246.4	1210.4	1456.8	0.9463	5.1621
16	752.79	0.001623		0.1693	255.9	1202.6	1458.5	0.9788	5.1385
18	803.66	0.001631		0.1590	265.4	1194.7	1460.0	1.0112	5.1151
20	857.12	0.001639		0.1494	274.9	1186.7	1461.5	1.0434	5.0920
22	913.27	0.001647		0.1405	284.4	1178.5	1462.9	1.0755	5.0692
24	972.19	0.001655		0.1322	294.0	1170.3	1464.3	1.1075	5.0467
26	1033.97	0.001663		0.1245	303.6	1162.0	1465.6	1.1394	5.0244
28	1098.71	0.001671		0.1173	313.2	1153.6	1466.8	1.1711	5.0023
30	1166.49	0.001680		0.1106	322.9	1145.0	1467.9	1.2028	4.9805
32	1237.41	0.001689		0.1044	332.6	1136.4	1469.0	1.2343	4.9589
34	1311.55	0.001698		0.0986	342.3	1127.6	1469.9	1.2656	4.9374

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